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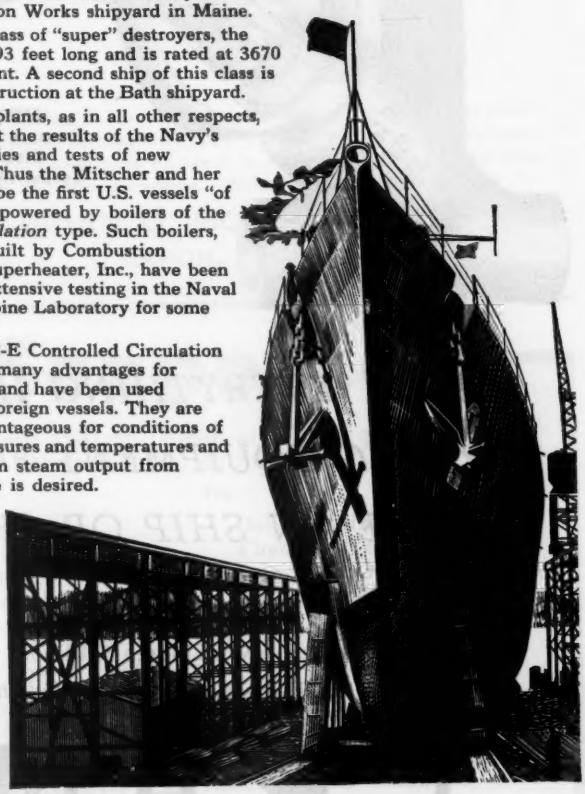

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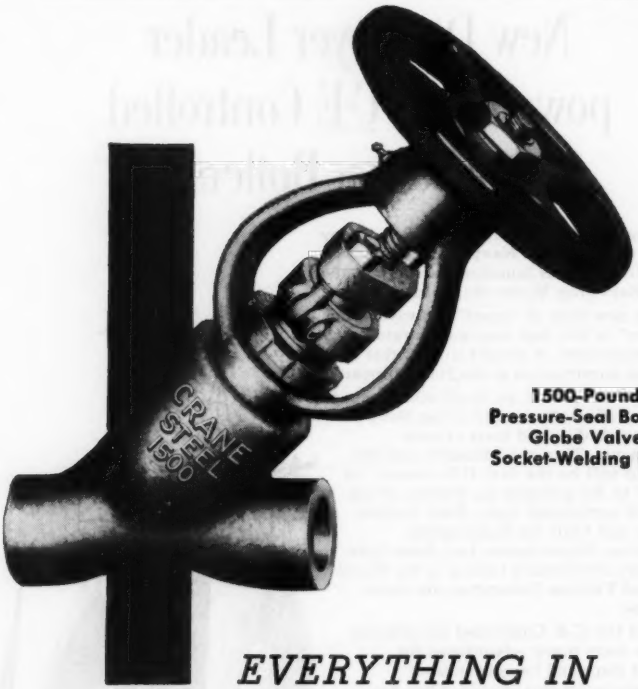

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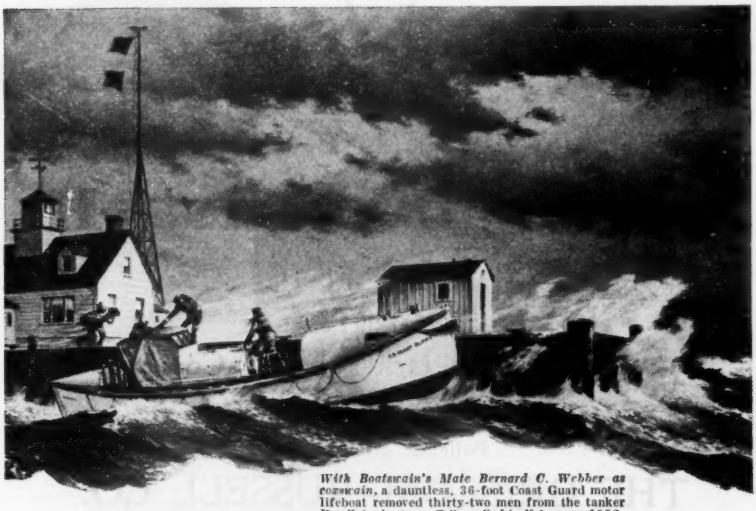

With Boatswain's Mate Bernard C. Webber as coxswain, a dauntless, 36-foot Coast Guard motor lifeboat removed thirty-two men from the tanker Pendleton's stern off Cape Cod in February, 1952.

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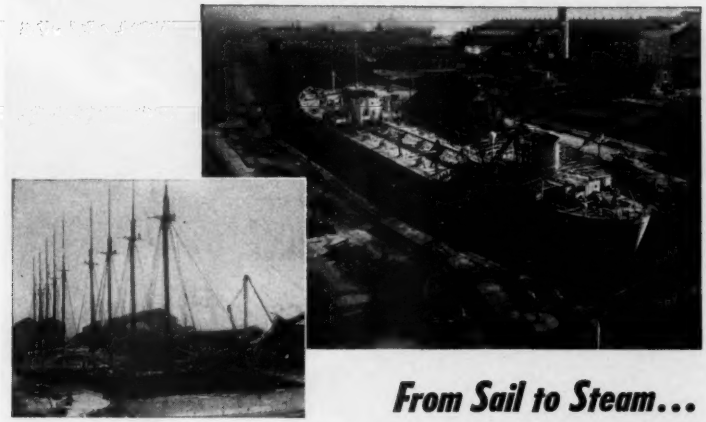

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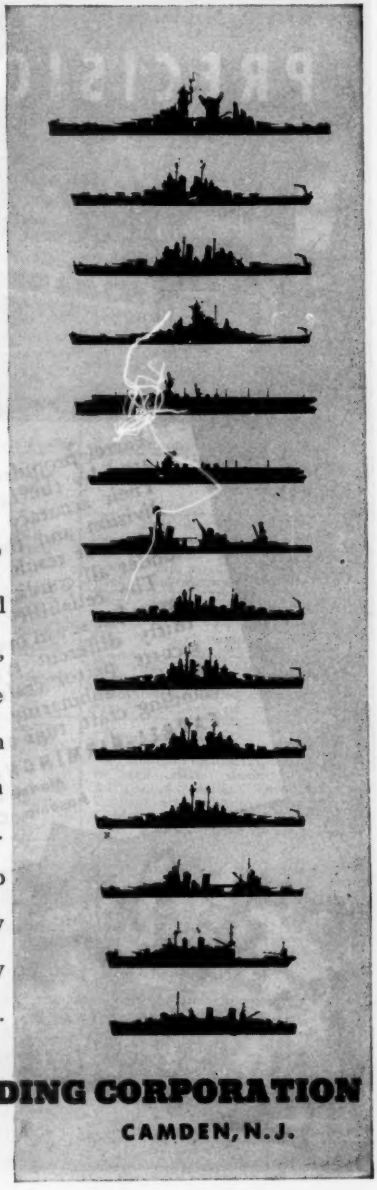

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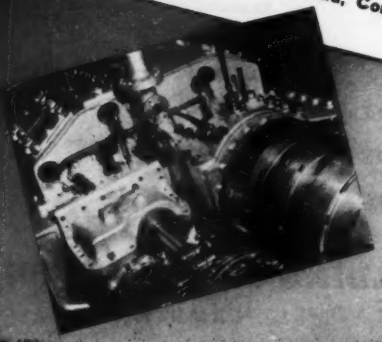

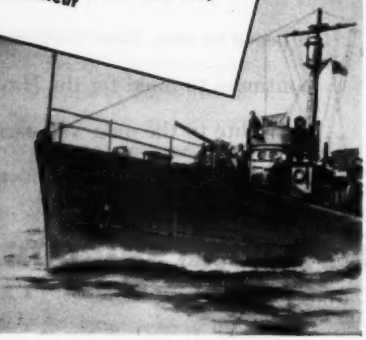

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Operations can be carried out with the work in crates or hung from hooks, utilizing an overhead rail and hoists. For large volume production, automatic equipment can be used to mechanize the line. Small parts can be treated in tumbling barrels.

"PERMADINE" DATA CHART

Type of coating	Zinc phosphate
Object of coating	Rust and corrosion prevention
Typical products treated	Nuts, bolts, screws, hardware items, tools, guns, cartridge clips, fire control instruments, metallic belt links, steel aircraft parts, certain steel projectiles and many other components
Scale of production	Large or small volume; large or small work
Method of application	Dip, or barrel tumbling
Equipment notes	Immersion tanks of suitable capacity. Cleaning and rinsing stages can be of mild steel. Coating stage can be of heavy mild steel or stainless steel.
Chemicals required	"Permadine" No. 1
Pre-cleaning methods	Any common degreasing method can be used. Alkali cleaning ("Ridosol"), Acid cleaning ("Deoxidine"), Emulsion-alkali cleaning ("Ridosol"-"Ridoline"); vapor degreasing, solvent wiping, etc., are examples.
Bath Temperature	190° - 210°F.
Coating time	20 - 30 minutes
Coating weight range Mgs./Sq. Ft.	1000 - 4000
Government Finish Specifications	U.S.A. 57-0-2C; Type II, Class B MIL-C-16232; Type II.

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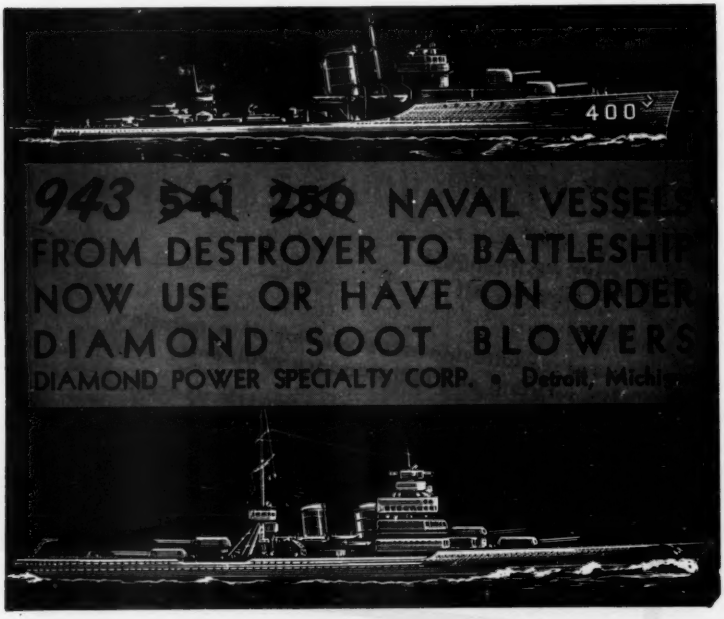
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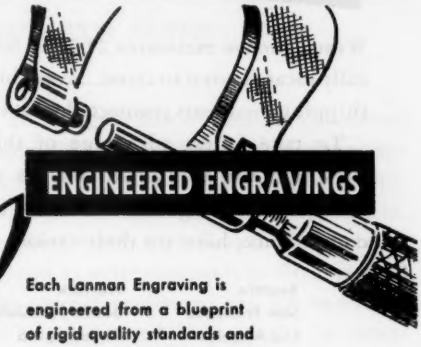

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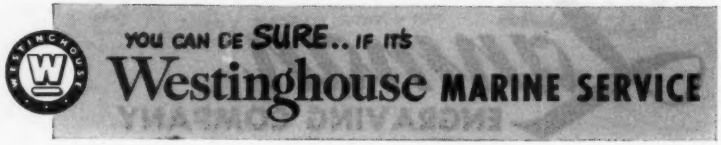

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SECRETARY'S NOTES

The Journal

There has been a very encouraging flow of manuscripts coming to the Editor's desk recently. We have a tendency to encourage authors to continue their pursuit of Naval Engineering by accepting papers which may be subject to some technical disagreement. This policy has the further effect, which we believe to be very beneficial, of encouraging disputants to write comments in disagreement. It will have been noted by careful readers of the JOURNAL that several technical disputes have appeared in its pages in recent issues. It is a firm opinion of the Editor that such disputes do more to accomplish the aims of the Society than any other device. No matter how faulty an article may be, comments and discussion by others and by the author present to the reader, not necessarily a final definitive answer, but all sides of a subject so that reasonably sound conclusions can be drawn and opinions can be formed.

In the interest of improving the JOURNAL as the organ of the Society for making good its existence, comment and discussion on the policy set forth in the preceding paragraph are urgently solicited. The JOURNAL should serve the members of the Society. Only by making their views known to the Editor, can he and the Council effectively manage the JOURNAL to serve this purpose.

The Annual Banquet

The final count on the 1952 annual banquet shows that 1578 members and guests were served. This number exceeds the previous high by 55. The popularity of this affair does not appear to be waning. The attendance is rapidly approaching the maximum capacity which Washington can accommodate. This maximum is believed to be 1632.

This year, for the first time, seats were assigned in order of receipt of applications. Those who were dissatisfied with their location may assume that the reason was that others had mailed earlier. It is probable that this precedence arrangement will be repeated in 1953. For the information of those who like to look ahead, the 1953 banquet has already been scheduled to be held in Washington at the Statler Hotel on

FRIDAY, 1 MAY 1953.

However, nothing will be gained by making application for the 1953 banquet before this can be done on the regular application form. As usual, these may be expected in members' mail about 1 March 1953. There will be no machinery for receiving and acting on applications before that date, and any which are received will be returned as in the past.

TIN CONSERVATION

MORRIS R. MACHOL

THE AUTHOR

is a graduate of Webb Institute of Naval Architecture and Marine Engineering. He has had wide and diversified experience since his graduation in 1903. In 1909 he presented the first paper on light weight reciprocating parts to the Society of Automotive Engineers. Has done much design and production work. During World War I he was with the Emergency Fleet Corporation where he developed a trend toward management and production engineering which he has followed in many important capacities. In 1947 he received a letter of commendation from UNRRA for his services in development of a Water Transportation Program for the Chinese Government. At the present time he is a Production Engineer in the U. S. Navy Bureau of Ships.

SUMMARY

Tin can be saved by:

1. Reducing the thickness of babbitt in babbitted bearings:
2. By changes in the design covering shape of bearings and methods of bonding.
3. By substituting babbitt metals with smaller tin content—for instance using lead base instead of tin base babbitts.
4. By reducing the percentage of tin in solder—and by adding very small quantities of other metals to compensate.
5. By specifying bronze compositions with smaller tin content where physical or corrosion-resistant characteristics will permit.

The above conclusions are supported by data and examples herein contained. The author of this brochure, as a consultant for the War Production Board during World War II, called on many manufacturers to discuss methods of conserving tin. Through the experiences of many organizations in different kinds of industry a number of practical ways to save tin were recorded.

This information has been collected, revised and is presented herewith.

CLASSIFICATION OF METHODS

The methods of saving tin automatically classify themselves into three main groups, the first, covering the use of tin in bearing or babbitt metal; the second, the use of tin in bronzes; the third, the use of tin in solder. The saving of tin in babbitts again divides itself into two parts; first, the substitution of lead base babbitts for tin base babbitts; and second, the redesign of bearings or the reconsideration of bearing designs in order to use less babbitt. This report will therefore be divided into five sections.

Section One will cover the replacing of tin base babbitts with lead base babbitts or other bearing materials.

Section Two will cover methods of design that would reduce the amount of babbitt used.

Section Three will discuss a few of the generally used brasses and bronzes that contain tin, their general characteristics and present uses, and the places where these alloys are necessary and where substitutions should be very carefully and perhaps grudgingly considered.

Section Four will discuss other brass and bronze alloys and other materials which contain no tin or little tin and which can be substituted for the tin-bearing bronzes mentioned in Section Three.

Section Five will comment on the use of tin in solder.

SECTION ONE

REPLACING TIN BASE BABBITTS WITH OTHER BEARING MATERIAL

Tin Base Babbitts

There are several tin base babbitts. The Maritime Commission uses either Post Motor-marine or A.S.T.M. #2, while the Navy has two or more of their own specifications. Practically all of the tin base babbitts contain between 80% and 93% tin. These tin base babbitts are generally considered to be the best performing and most desirable bearing metals obtainable for almost all purposes, and if it were not for the scarcity of tin there would have been no question of substituting any other metal for high tin base babbitt except under special or unusual circumstances.

Lead Base Babbitt, Antimony—Arsenic Compounds

There are a number of compounds using antimony and arsenic which seem to have overcome weak points previously evident in lead base babbitts, namely, hardness inferiority to the tin base babbitts at elevated temperatures, corrosion resistance and erratic performance. Two of these alloys are as follows:

(a) 12½% antimony, 3% arsenic, ¾% tin, balance lead

(b) 15% antimony, 1% arsenic, 1% copper, 1% tin, balance lead.

These babbitts must be poured at temperatures considerably higher than required for tin base babbitts, and the technique of handling them is slightly different and requires some patient attention.

The lead base babbitts are proving so satisfactory that they continue to be used in many places.

Other Lead Base Babbitts

There are many other lead base babbitts, some containing small quantities of silver and some having alkali-hardened lead. Many people are experimenting with various lead base babbitts and as the tin shortage becomes more acute there will probably be more and more

lead base babbitts substituted for the tin base babbitts. The alkali-hardened lead base babbitts contain small quantities of calcium. There are two disadvantages with this alloy. (1) It has low resistance to corrosion in the presence of additive oils. (2) If the metal is melted for recasting, the calcium disappears. Therefore, this type of babbitt cannot be melted out and poured back into a bearing like other babbitt metals. A small amount of indium alloyed with lead provides hardness and corrosion resistance without destroying imbedability, and thus produces an excellent bearing metal. Cadmium is another metal that added to lead in small quantities hardens it and makes it an excellent bearing metal. Its disadvantages are cost and the fact that Cadmium treated lead is very susceptible to corrosion.

Substitution

It is obviously desirable wherever possible to save tin by substituting a lead base babbitt for a tin base babbitt. There are some places, however, where such substitutions must be approached very carefully, for instance, the thrust bearings, turbine bearings, propeller shaft bearings, or reduction gear bearings of a battleship or other combat vessels are vital points and failure at a critical time might easily become a major catastrophe. Therefore, those responsible for designs or specifications should be most reluctant to depart from either designs or from the use of materials which through years of service have shown themselves to be reliable. The fact that these points are vital should not preclude the consideration of changes to save tin or other critical metals. It merely means that the designer must be very careful and very sure before authorizing the change.

Phenolic Impregnated Fabric Bearings

A number of companies in the United States are producing bearings made of phenolic plastic impregnated fabric. There are many trade names, among them are Textolite, Micarta, Ryetex, Gatke, etc. These materials can be used in place of anti-friction metals where conditions permit the use of running water or of solvent oils as a lubricant. This material has excellent anti-friction qualities, but practically no thermal conductivity, whatever. In a place where it is not possible to keep it both lubricated and cooled with water, the bearing surface as well as the journal would soon get extremely hot and failure would result. The material has no imbedability and bits of grit or sand if present, would not imbed themselves as they would in a lead base babbitt bearing.

SECTION TWO

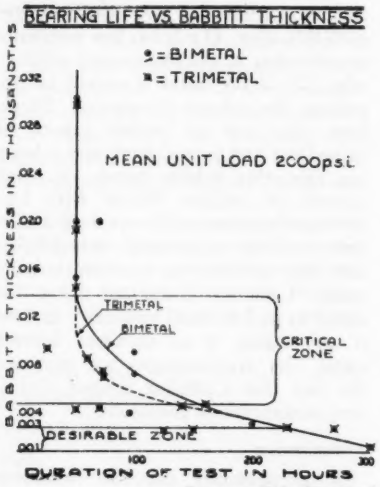
DESIGN METHODS—TO REDUCE USE OF BABBITT

The Use of Thick Babbi't Bearings

For many years, particularly in heavy service, it has been the practice to use a fairly good thickness of babbitted bearings. From 3/8 to 3/4 of an inch of babbitt has been quite common and some bearings run up to even greater thicknesses. The theory on which engineers provided a thick babbitt bearing was probably that first, the babbitt would wear and the thick layer provided for plenty of wear; and second, that babbitt was a soft metal and would act as a cushion, thereby absorbing shock. The writer would like to submit, based on what has been done in the automotive industry, that both of these assumptions should be definitely questioned. The standard automotive crank shaft bearing

is a hard steel back on which about 0.005" of babbitt has been applied and there is seldom any question of the failure of these babbitt bearings from wear. One airplane engine has a main bearing consisting of a steel back on which 0.015" of silver has been deposited and in addition 0.001" of lead. The lead takes care of small inequalities in the journal. Another airplane engine has a steel backed bearing with a thin layer of copper-lead metal, on top of which is an electrolitic deposit of about 0.001" of lead. An English engine also has a copper-lead steel back bearing coated with as little as 1/10 of one thousandth inch of lead. These bearings with pressures running as high at times as 12,000 to 15,000 p.s.i. and up to 5000 R.P.M. show very little, if any, evidence of wear. It should be noted that with a bearing of this type there is first no cushioning effect; and second, there is minimum resistance to the transmission of heat. A thick babbitt bearing, if it acts as a cushion, may be destroying itself due to the fatigue of the metal, by breaking up the adhesion between the crystals of the metal, and thus destroying its inter-crystalline structure. Also, if in such a bearing there is a tendency to generate heat, the babbitt, being a very poor conductor of heat is much more easily destroyed if it has a thick section. Obviously a section of lead only 1/1000 of an inch thick offers practically no resistance to the transfer of heat. The fact that these aviation engines with their high speed and high bearing pressures and no cushioning show practically no wear, would seem to prove that a thin babbitted bearing will have a much longer life than a thick one. The conclusion, that the thinner the layer of babbitt is made, the longer the life of the bearing will be, is inescapable.

Tests Showing Effect of Thickness of Bearing on Life



Courtesy of the Cleveland Graphite Bronze Company

The Cleveland Graphite Bronze Co. in cooperation with the Chrysler Corp. made a series of tests on steel backed and also on bronze backed babbitted bearings as well as some tests on trimetal bearings (a layer of bronze on a steel back and babbitt on the bronze). A copy of the chart showing the result of these tests is here shown. From 0.032 down to 0.015 there was no difference in the life of the bearing. As the thickness was reduced the life increased, and below point 0.012" thick the rate of increase was very rapid.

Reluctance to Use Steel Backed Thin Babbitted Bearing

There is an understandable reluctance on the part of any conservative engineer to substitute a very thin babbitted bearing in a place where a bearing with a thick layer of babbitt has given satisfactory service for years. This is based on

his fear that the babbitt will give out and that contact of the steel shaft on a ferrous liner or pedestal will score and destroy the shaft. It is obviously good practice to ruin the bearing metal which can easily be replaced by rebabbitting rather than to ruin the shaft itself.

Use of Bronze Backed Babbitted Bearings

In order to save tin, a babbitt bearing of perhaps ½" thickness might be replaced by a bronze backed babbitt bearing with a section of 1/16" or less of babbitt. This would not necessarily mean using any babbitt different from the high tin base or other babbitt that had previously been used. Nor would it provide any different bearing surface that had previously been provided. In case, for any reason, the babbitt should fail there would be the soft bronze backing that would be a good bearing metal "on its own." And a shaft would continue to run without scoring in a bronze bearing at temperatures considerably above the melting point of babbitt. A conservative engineer who would be very reluctant to accept the responsibility of using a thinner section of babbitt on a ferrous back would frequently consider it safe to do so if he had the softer bronze back to depend upon in case anything happened to the babbitt.

Suggested Composition for Bronze Back

A bronze containing from 70% to 85% copper and from 30% to 15% lead makes an excellent anti-friction bronze. It does not have a great deal of tensile strength but it is plastic and will accommodate itself to considerable distortion. The higher percentage of lead gives better anti-friction qualities to the metal and also increases the difficulty of getting castings without lead segregation.

(See later paragraphs for further remarks on these bronzes) and for their casting characteristics.

Anchoring

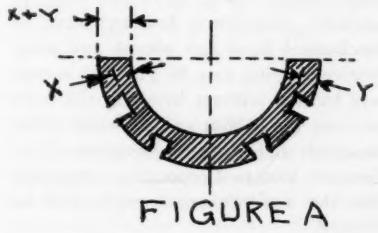


FIGURE A

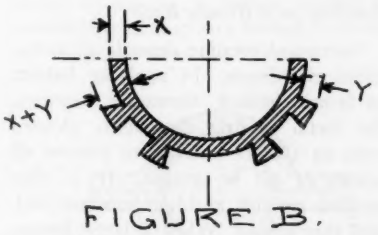


FIGURE B.

In many places I have found the type of anchor shown in figure A. The thickness of the anchor "Y" is usually about the same as the thickness of the babbitt "X". In this type of anchor for all practical purposes the thickness of the babbitt would be the thickness shown at "X", because if this thickness of babbitt were worn away, the journal would contact the harder back. The thickness "Y" is obviously of no value, as far as bearing service is concerned. But to all intents and purposes, as regards the amount of metal used, the thickness of the babbitt used would be twice "X". The type of bearing, shown in Figure B would have the same thickness of bearing metal but would use considerably less bearing metal in the anchors than the type of bearing shown in Figure A.

Anchoring Unnecessary With Proper Bond

If the babbitt is properly bonded either to a ferrous back or to a bronze back there should usually be no necessity for anchors. Automotive bearings have no mechanical bond and almost any automotive bearing can be put into a vise and twisted without breaking the bond between the babbitt and the steel. (The Research Laboratories Division of the General Motors Corporation suggested that the word "almost" might well be omitted.)

Bonding on a Bronze Back

Successful bonding depends on meticulous cleanliness. In applying babbitt to bronze backed automotive bearings, the metal is first thoroughly cleaned with an alkaline solution to remove all traces of oil or grease. It is then pickled, perhaps in dilute muriatic acid, and then tinned. When a large bronze shell is to be babbitted, a shell such as is used on a railroad journal bearing, a recommended procedure would be as follows: The bronze shell is rough turned in a lathe and immediately coated with a flux such as any of the commercial solder pastes or compounds. This protects the surface from oxidation until it can be tinned either with pure tin, with tin-lead solder or with one of the acceptable commercial tinning compounds. If the babbitt can be immediately poured a good bond will be obtained. If not, the tin will very quickly acquire a yellowish hue due to oxidation. In this case, before tinning, treatment with a salamoniac will remove the oxide. The salamoniac should then be washed off, preferably with a steam hose and the babbitt immediately poured. If the babbitt can be centrifugally cast so much the better. Even after all these precautions have been taken the bond can be ruined by improper chilling. If chilled from the inner surface the bearing surface shrinks away from the backing while babbitt is still fluid resulting in porousity and cleavage. The bearing should ALWAYS be chilled from the back.

Bonding an a Malleable or Cast Iron Back

Time was when it was considered impossible to bond bearing metal on to a cast or malleable iron back. The free carbon or graphite on the iron surface is what gives iron a greasy feeling and prevents a satisfactory bond.

The next improvement was to remove much if not most of this free carbon by grit blasting. (Not sand or shot blasting which would pound the carbon in)—immediate wire brushing and fluxing had to follow. This however did not remove all the carbon. Then the Kolene process came along. This removes all the surface carbon. The iron back is placed in a molten salt bath at a temperature of 760° to 800°. This not only takes out the free graphite but also tends to de-carburize the surface; it also oxidizes the surface and this oxide must be removed. So the iron is rinsed and dipped in a hot alkaline bath, the oxide is thus dissolved off the surface. Another rinse and a dip, into a 10% solution of HCl will neutralize any remaining oxide. Immediate coating with flux prepares the surface for babbitting. Copper plating or brazing of the surface (preferably in a hydrogen furnace) will insure a good bond.

Bonding on a Steel Back

A similar procedure is effective in obtaining a bond on a steel back. If any oil has been used as a cutting com-

pound the surface should be washed in an alkaline solution and preferably then pickled. If this is not done even small quantities of oil may collect on the surface of the pickling solution and redeposit on the bearing when it is removed from the bath. The point is that absolute cleanliness is necessary before tinning in order to secure perfect bond. After the surface is tinned the babbitt should be either poured at once or the tin oxide removed with salamoniac. The Kolene process described above is also applicable to steel backs.

Symmetry Not Essential

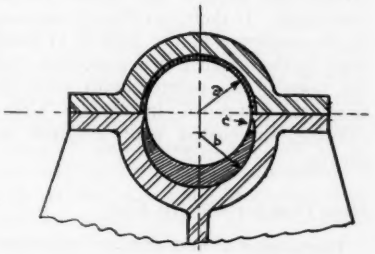
It is quite natural for an engineer designing a bearing to make it symmetrical top and bottom. The forces on a bearing can be resolved in a single direction. Often this resolved force will be a small amount, perhaps ten to fifteen degrees off bottom center when the shaft is revolving in one direction. Often when the machinery is reversed such as in the case of reduction gear bearings on a marine propulsion job, the direction of the resolved force will be about the same angle on the other side of the center and not 180 degrees away. Obviously, in such cases or in cases where the bearing merely carries a heavy weight, the pressure is always in one direction. Therefore, in such cases, especially where centrifugal casting of the babbitt is possible, it is entirely practical to use a very thin film of babbitt, perhaps $\frac{1}{64}$" for the non-pressure half of the bearing. (See figure in opposite column.) To do this it is only necessary to put two caps together and bore the hole $\frac{1}{32}$" larger than the shaft diameter. Two pressure halves can often be bored together in the same way.

Special Conditions Not Appropriate to Bronze Backed Bearings

The reason or perhaps the excuse for a bronze backed bearing is the idea that if the babbitt is destroyed or wears through, a bearing bronze is provided to insure against steel on steel. In the case of extreme speeds such as are attained in high speed turbines the peripheral speed should be carefully considered because if it is too high, bronze is very prone to seize and score the shaft.

Eccentric Babbitt Section

Assuming that a designing engineer decided on a certain necessary thickness of babbitt, let us say one-half inch, and assuming that he considered it desirable to save tin by using very thin babbitt lining in the cap, a further method of eliminating unnecessary babbitt is shown



in the illustration. In boring the lower half of the bearing, instead of boring it concentric with the journal, drop the center from center *a*, perhaps $\frac{7}{16}$ to center *b*, and increase the radius $\frac{1}{16}$". This gives the same half-inch thickness of babbitt at the bottom of bearing, but at the sides of the bearing the thickness is very materially reduced. If necessary a hole can be drilled in one side "*c*," to be used as a gate in pouring the babbitt. This moon shaped section frequently saves considerable bearing metal without in any way impairing the effectiveness of the bearing.

Re-babbitting

In a shop where the babbit is centrifugally cast there is no difficulty in

getting as thin a layer of babbitt as desired. Where the babbitt is cast around a mandril there should be no difficulty in pouring the babbitt if both the mandril and the shell are heated to a proper temperature. It is conceivable however that it might be difficult to pour in a situation that would be presented for instance, in re-babbitting a long shaft bearing, where it was impractical to take out the shaft and use a mandril, and where it would be dangerous to attempt to heat the shaft with a flame. It is suggested that in such a case, especially if superheated steam were available, both the shell and the shaft should be heated to 300 or 400 degrees or more. This works very effectively, as far as pouring the babbitt is concerned, and without any danger to the shaft itself. If there is difficulty because of the tendency of the babbitt to bond itself to the mandril, it is suggested that the mandril be painted with colloidal graphite suspended in carbon tetrachloride, or coated with graphite or grease.

Heat Transfer and Air Gaps

The subject of heat transfer especially in bearings is one that should be given consideration. Whenever a journal or shaft is rotated in a bearing heat is generated, and unless provision is made to carry the heat away as rapidly as it is generated, the temperature builds up. Of course, the generation of heat increases the temperature. This in turn increases the flow of heat until a balance is reached. The generation of heat can be controlled by proper selection of antifriction metal and by proper lubrication. The selection of materials for the backings should not be made without giving some consideration to the ability of the metal to carry away the heat, that is, its thermal conductivity. In this connection, it should be remembered that an air gap is a very effective insulator and therefore a barrier to heat transfer. In building Liberty motors in first World War, an attempt was made to use an aluminum cylinder with a ferrous liner. Both the liner and the cylinder were carefully machined and the liner was pressed into place. In spite of the close fit, this air gap offered such a high resistance to heat transfer as to make its design ineffective. A steel bar with a crack that can only be seen in a microscope can be heated to a cherry red on one side of the crack and remain black on the other side. Heat can only be transferred freely from one metal to another if there is a thermal bond, a soldered, brazed, welded or tinned joint. In a situation where an air gap is inescapable the problem of heat transfer can be met with a heat drain. One end of a piece of copper wire or cable can be soldered to the bearing and the other end to a member whose area and position enables it to disperse the heat by radiation. In this way the air gap can be short-circuited and the bearing kept cool.

Spraying

I have discussed the subject of spraying babbitt on to either a ferrous or non-ferrous back with several people. They all agree that this method produces an excellent bond but each one informs me that the spraying seems to embrittle the metal to a point that materially reduces its value as a bearing metal.

Facing a Lead Base Babbitt With a Tin Base Babbitt

Obviously it is very desirable to save tin by substituting a lead base babbitt for a tin base babbitt. Two of the principal advantages of the tin base babbitt are that it is harder and has a higher resistance against corrosion. Where a fairly thick layer of babbitt is desired it is suggested that consideration be given to pouring the lead base babbitt first and then applying a facing or thin coat of tin base babbitt. This would

provide the hard, corrosion-resistant wearing surface desired. There would be no difficulty in getting a satisfactory bond between the lead base and the tin base babbitts.

Length of Bearings

The unit pressure of a bearing can be reduced by increasing its length. Obviously a bearing two diameters long will carry half the p.s.i. of the same bearing one diameter in length. This however is usually not an advantage. The bending (distortion) of the shaft under stress causes bell-mouthing. Especially where clearances are in tenths of a thousand of an inch, length is a distinct disadvantage in a bearing. Generally speaking, a bearing will give better service if it is not over one diameter in length, even though this may increase the unit pressure. As in many installations ⅔ or ¾ of the diameter will give the longest bearing life. Even a minor load on a shaft will cause a minute distortion.

Oil Grooves

The shape, depth, width, length and position of oil grooves is a matter of utmost importance. The automotive industry has spent much time and money in experiments to determine the best design for oil grooves under different conditions. The oil pressure that is built up between the journal and the surface of the bearing creates the film that prevents wear and trouble. This pressure is high at the middle of the bearing, and runs down to nothing at its ends, and obviously the pressure is higher at the point of highest load which is usually the top or bottom of the bearing. The edges of the oil grooves should be rounded and not sharp. And wedge-shaped spaces at the sides help the shaft to pick up the oil and force it into the higher pressure areas. Any oil groove that passes through the area of high pressure materially reduces the efficiency of the bearing. The purpose of the oil groove is to lead the oil to the point at which the journal can pick it up and carry it around. A groove in the low pressure area parallel to the shaft will act as a trap or scraper groove and will pick up most of the foreign matter, grit, etc., that might otherwise cause trouble. Where it is necessary to run a groove circumferentially around a bearing it creates two pressure areas instead of one. Both study and experimentation are worthwhile in determining how and where to place the oil groove.

SECTION THREE

TIN BEARING BRONZES

General Character of Following Comments

The paragraphs comprising sections three and four are to be considered merely comments on various metals. These comments are intended to be only of a general nature, referring to classes of alloys and not to exact or definite specification. The American Society For Testing Materials has published numerous specifications and has made recommendations for emergency alternates during the first World War. Engineers and designers in considering the following paragraphs should always refer to definite specifications published by either the A.S.T.M., branches of the Government, or other recognized authorities.

Gun Metal (Composition "G")

Gun metal is a strong, tough, readily cast and readily machinable bronze alloy. The traditional analysis is 88% copper, 10% tin and 2% zinc. The analysis that should be most frequently used is 88%, 8% and 4%. Generally both of these metals will give an ultimate tensile strength of 40,000 to 55,000 p.s.i., a yield point of 18,000 to 26,000 p.s.i., and an elongation of 18-55%. An outstanding characteristic of this metal is its corrosion resistance, especially in the presence of still or running salt water. The metal is soft, having a Brinell hardness of about 60 to 70. Gun metal has a very low thermal conductivity, 0.0150 as compared with 0.923 for pure copper. Because of its resistance to salt water and the ease with which it can be cast and machined, this alloy has been very extensively used. Every effort should be made to discontinue its use because of the amount of tin it contains. For most marine and naval uses composition "M" can be substituted where resistance to salt water is required. The only place where composition "G" should be used is where resistance to salt water must be provided and where the design is such that the additional strength of "G" over "M" is necessary. Occasionally, a very intricate casting that might be impossible to cast in manganese bronze or silicon bronze might require composition "G" or "M". This metal is extensively used for bearings in naval ordnance.

Composition "M"

Another analysis that shows great resistance to corrosion in salt water is 88% copper, 6% tin, 1½% lead, and 4-4½% zinc. This will have an ultimate tensile strength of 34,000 to 50,000 p.s.i. A yield point of 15,000 to 21,000 p.s.i., and an elongation that will run from 12% to 35%. This alloy is extremely easy to handle in the foundry and very easily machined. It is a very desirable alloy to use in place of Composition "G". This metal is comparatively soft. Its Brinell is 60-72, thermal conductivity 0.195, or somewhat better than Composition "G". The outstanding feature of Composition "M" is that it maintains its stiffness and its physical characteristics at fairly high temperatures and can be used for pressure tight castings where high temperatures are encountered, such as in valves for superheated steam.

Hydraulic Bronze

Another very generally used specification is 85% copper, 5% tin, 5% lead and 5% zinc. This alloy will show an ultimate tensile strength of about 30,000 to 45,000 p.s.i., a yield of about 14,000 to 24,000 p.s.i., and an elongation of 15% to 35%. The Brinell hardness is 55-65. This alloy is extremely easy to handle both in the foundry and machine shop. It has an excellent corrosion resistance to salt water and other elements. It is also a dense material and suitable for pressure-tight castings, such as in high pressure water valves and connections. It, however, is not nearly so good as composition "M" at high temperatures.

5% Tin Required in Salt Water

With the exception of the manganese bronzes it can be generally stated that a minimum of 5% of tin is necessary in a copper alloy to insure satisfactory corrosion resistance in salt water.

Composition "H"

This is an alloy containing about 83% copper, 13% tin, 3% zinc, 1% lead. It has excellent physical characteristics. This alloy will give an ultimate tensile strength of 28,000 to 32,000 p.s.i., and from 10-20% elongation. It is an excellent bearing metal where slow speeds and heavy pressures are encountered. Reducing the zinc will increase its anti-

friction qualities. This is a close grained, rather brittle metal and particularly well suited for pressure-tight castings but not at high temperatures. Its Brinell hardness is 60 to 70. When nickel is available—1½% nickel replacing 1½% tin materially increases both strength and bearing qualities.

Phosphor Bronze

There are two general kinds of phosphor bronze, cast and wrought. The base of cast phosphor bronze is 80% copper, 10% tin and 10% lead. A modification of this type is 84% copper, 8% tin and 8% lead. The usual wrought phosphor bronze is 5%, 8% or 10% tin, the balance copper. The addition of enough phosphorus is made to deoxidize the metal, and leave a residual of from one-quarter to one-half percent of phosphorus. This hardens the alloy, somewhat, embrittles it, and cuts down its elongation. The hardness of phosphor bronze makes it a very good wearing metal available for uses such as gears where a low speed, high pressure bearing is desired. Cast phosphor bronze will show physical characteristics somewhat below Composition "G". A phospher bronze containing 96% copper, 4% tin and 0.3% phosphorus is a good alloy. Some phosphor bronzes can be rolled or drawn and make excellent springs where ferrous springs would be subject to corrosion. An ultimate tensile strength of 130,000 p.s.i. can be obtained on thin sections. Brinell hardness approaching 300 are obtainable. Reduction of the tin content makes it easier to forge or roll the metal. The higher the tin content the greater is the strength obtainable.

Tin-Bearing Bronzes Should Be Avoided

There are, of course, many places where the peculiar characteristics of tin-bearing bronzes make their use essential, but there are also many places where these bronzes have been generally used and where other bronzes that do not contain tin can be used to replace them. Engineers, in writing specifications for the duration of the emergency, should be very careful to use the tin-bearing bronzes only where there is no acceptable substitute.

SECTION FOUR

BRASSES AND BRONZES WITH LITTLE OR NO TIN AND FERROUS METALS

Silicon Bronze

Copper silicon alloys containing up to 5% silicon are very acceptable substitutes for tin-bearing bronzes in some places. A number of copper silicon alloys have been patented, and while some of the patents have expired, this situation should be checked where copper silicon alloys are specified. Although they are not covered by many accepted standard specifications the use of 5% to 16% zinc combined with manganese up to 1½%, silicon up to 5%, iron up to 2½%, and tin up to 2% will develop physical values very much in excess of the tin bronzes. Tensile strengths of 50,000 to 75,000 lbs. with 20% elongation are not uncommon. The silicon alloys are difficult to handle in the foundry and not as easily machined as the tin alloys. The addition of a very small amount of lead will facilitate both machining and casting, but will reduce the physicals. Silicon bronzes require that the foundry be much more liberal in the use of risers; at least one riser for every boss on the casting is necessary. Gates and sprues must usually be larger, which of course, decreases the yield per ton of cast metal. It is possible to produce intricate castings of sili-

con bronzes if the risers and gates are carefully planned and are adequate in number and size. Chills must be used freely. The shrink characteristics are unpredictable, and on any intricate casting it is usually necessary to make and study experimental castings. Silicon bronzes in general are an excellent substitute for bronzes containing tin, and frequently, because the iron so increases the physical characteristics, sections can be cut down and the weight of the casting reduced and copper saved. Silicon bronzes are hard, the Brinell runs from 100-130. It should be remembered that the presence of silicon in a bronze alloy ruins its qualities as a bearing metal, as well as its thermal and electrical conductivity. The thermal conductivity of silicon bronzes run from 0.08 to 0.11. Silicon bronze will not stand up in salt water.

Manganese Bronze

Commercial manganese bronze should perhaps strictly be called manganese brass. Its base is 60% copper and 40% zinc. The addition of 3% or 4% of manganese toughens the metal and deoxidizes it. In some manganese bronzes only a trace of manganese is left in the metal. A small amount of iron gives it considerable strength. There is no difficulty getting an ultimate of tensile strength from 65,000 to 75,000 p.s.i. The yield point is fairly low, 25,000 to 40,000 p.s.i., and the elongation varies from 8% to 35% according to the strength; 20% to 25% is normal. 60% copper, 37% zinc, 3% manganese is a standard bronze for marine propeller use. Manganese bronze is not easy to handle in the foundry, and is a little difficult to machine. Lead up to 3% does not too greatly impair physicals, makes it somewhat easier to handle in the foundry, increases machinability, and more important, makes copper scrap usable in producing the alloy. Increasing the copper to 66⅔% and decreasing the zinc to 33⅓% makes the metal much easier to handle both in the foundry and in the shop, but reduces the physical properties. Manganese bronze is corrosion resistant in salt water and is an excellent substitute for tin-bearing bronze where salt water, salt water spray or other corrosive conditions are encountered. With a little iron, tensile strengths up to 130,000 p.s.i., and Brinells of 150 to 200 are easily obtainable. The metal is usually porous and should not be used for pressure-tight castings. An interesting feature of manganese bronze is that its thermal conductivity is 0.241, nearly twice that of other bronzes. Manganese bronzes have the advantage of great strength and ductility, resistance to corrosion, and the additional advantage that they conserve both tin and copper.

Navy Brass

A typical navy brass for sand castings is 60% to 67% copper, sometimes 1% to 0.3% lead, 1% tin and the balance zinc. In general, navy brass has physical characteristics about the same as hydraulic bronze. (Eighty-five and Three Fives.) The metal is easy to cast, it is easy to machine, has the advantage that it conserves copper as well as tin, and is a very desirable metal to use wherever just a bronze casting is required. It offers fair resistance to salt water and can be used where there is occasional salt spray or even occasional drenching with salt water. This metal should not be used for bearings.

Copper-Lead Bronzes

Alloys from 70% to 85% copper and from 30% to 15% of lead have a very high anti-friction quality. Sixty percent copper and forty percent lead is even better, but much more difficult to cast. These bronzes are not stable. The higher the percentage of lead the more unstable the metal. The tendency of the lead will be to segregate, and segregated areas of lead on the surface of a bearing are very liable to seize and score the shaft.

And segregation of the lead materially decreases the load carrying capacity and the strength of the bearing. In order to prevent the segregation of the lead it is necessary to keep the metal agitated in the pot and to keep the temperature down to 2100°F., and preferably closer to 2000°F. Quick chilling will help greatly in this respect.

If the mould is carefully designed, and if chills are used in order to assure that the casting will be cooled uniformly and quickly, the danger of segregation is materially reduced. This metal is not very strong, tensile strengths of approximately 20,000 p.s.i. are obtainable. The metal is hot short at about 600°, but not quite so hot short as it is if tin is added. The addition of small quantities of tin, while it gives added strength, greatly impairs anti-friction quality of the metal. The addition of zinc to a bronze of this sort will also definitely increase its strength. In addition, zinc will make it very much more susceptible to corrosion from impure or lard base oils. These copper-lead bronzes have practically no elongation, but they are softer and more plastic than the bronzes containing tin.

Bearing Characteristics of Copper-Lead Bronze

Generally speaking, a bronze containing 40% lead and 60% copper is the best bearing metal of this type. As the lead is decreased and the copper increased the non-score quality is decreased. The addition of tin, and in fact any other metal that adds hardness and strength, will decrease the value of the metal as a bearing metal. For heavily loaded main bearings or for connecting rod bearings of internal combustion engine the copper-lead bronzes are excellent. For high speed bushings of small diameter where lubrication is not well maintained, the addition of a small amount of tin as a substitute for some of the lead is desirable. For moderate speeds with small bushings a small amount of zinc is helpful. For instance, an alloy containing 83%-88% copper, 4½% to 7% tin, 3½% zinc, 4%-7% lead, gives excellent service. For heavily loaded bearings where the lubrication is well maintained and where the motion is slow or oscillating, such as would occur in a heavily loaded wrist pin, stronger metal such as 88% copper, 10% tin and 2% zinc make excellent bearings.

Copper-Aluminum Alloys

Generally speaking, the addition of aluminum and iron to copper gives considerable added strength and hardness. Aluminum bronzes contain from 5%-11% of aluminum, from 1%-5% of iron and the balance copper. The higher percentages of aluminum and iron give high physical characteristics. As cast, this metal will run up as high as 80,000 p.s.i. ultimate tensile and when heat treated or forged it runs up higher. It is not easy to cast. It has a Brinell hardness of 100 to 200. This metal shows high resistance to wear. The outstanding feature of ferric aluminum bronzes is that they retain their strength at high temperatures. Scarcity of aluminum is not an insurmountable barrier to the use of aluminum bronze since secondary aluminum can be used.

Phenolic Varnish

Phenolic varnish can be used with good results to counteract porosity and to prevent corrosion in bronze castings. When the metal is warmed enough to dry it thoroughly and sprayed with a low oil base phenolic varnish, the varnish will be absorbed into the pores of the cooling metal. If this is done just before the final machining the varnish that gets into the pores will serve to make what might often be a porous casting tight against lower pressures, and sometimes such a use of phenolic varnish will help materially to prevent corrosion.

Malleable Iron

Malleable iron is not nearly as critical a material as copper, and it not only can often be substituted for bronze but should be so substituted wherever possible. Its ductility and resistance to shock compares favorably to bronze. The normal physicals are approximately in the same class as those of composition "G". Pearlitic malleable irons with their arrested annealing will develop a strength from 50,000 to 70,000 p.s.i. Malleable iron has a thermal conductivity of 0.15. Malleable iron is so much less critical than the non-ferrous metals that engineers should give this metal every possible consideration.

Cast Iron

The thermal conductivity of cast iron is 0.13. There are many grades of cast iron having a wide range of physical characteristics. Cast iron is brittle and should never be used where it is likely to be exposed to the shock of explosions. One type of cast iron is Ductile Iron controlled by a patent. It is made with nickel and magnesium—practically none of the magnesium and up to 4% or 5% of the nickel remains in the finished casting. The resulting iron has a pearlitic structure. The carbon is retained in nodular form. Tensile strengths of 65,000 to 125,000 lbs. are obtainable against a maximum of about 65,000 for regular iron. Elongation of 5% to 20% is a characteristic against no elongation for ordinary iron. There are many places where ductile iron can be profitably substituted for cast steel.

Porous Bearings of Sintered Bronze or Iron

A number of companies are manufacturing this type of bearing, particularly for smaller bearings. The bearing is made by sintering powdered metal in a die in such a way as to produce a very porous mass. It is said to absorb enough oil to take care of lubrication for a reasonable length of time if the regular oil feed should fail. The use of this type of bearing metal made from a bronze containing no tin, or from iron, can be very economical as far as the use of critical materials is concerned, and should receive the consideration of engineers both for manufactured products and as replacement bearings.

Ferrous Back Babbitted Bearings

Tin and copper can be saved by substituting for a thick tin base babbitt bearing, a malleable or forged steel back into which has been cast a moon-shaped segment of bronze, preferably of the copper-lead type. The bronze surface can then be babbitted with a thin coat of whatever kind of babbitt is best suited for the service required. Such a bearing has an excellent babbitt surface for the journal to run on, and in case anything does happen to the babbitt, it has a bronze back to fall back on, a bronze back which is in itself a good bearing metal. The steel or malleable back would be lighter, cheaper and stronger than the bronze. If bronze is poured into a bearing, heated to a temperature near the melting point of the bronze, an excellent thermal bond between the bronze and the ferrous metal is easily obtainable. Mechanical anchors can also be used if desired. In the case of malleable, the bronze should be poured before the castings are annealed, then the casting can be annealed with the bronze insert; otherwise, the malleable will revert to cast iron. In such cases no more than 15% lead should be used and whatever lead is sweated out on the surface during the annealing should be scraped or machined off.

SECTION FIVE

SOLDER

Solder With Lower Tin Content

Solder uses a considerable amount of tin. In normal times 50% tin and 50% lead is a very popular solder ratio. Reducing the tin and increasing the lead content is a conservation measure.

Difficulties

Many people have had poor results with tests of solder having a low tin content. When imperfect joints resulted, they reverted to the 50-50 solder. It is probable that much of this difficulty stems from a lack of understanding of why the parts should be kept immobilized for a longer time.

Requirements For a Good Joint

A satisfactory joint requires that the solder shall be liquid—that the parts to be soldered shall be at least as hot as the liquid solder and that the parts be prevented from moving until the solder is in the solid state.

Solid, Plastic and Liquid States

Tin-lead alloys containing from 10% to 98% tin become solid at 361°F. An alloy with 62% tin goes directly from the solid to the liquid state at 361°F. Any other tin content means a plastic state between the solid state and the liquid state. As the tin content increases from 62% to 98% the liquid (melting) point gradually increases to 440°F. As the tin content decreases down to 10% the melting point increases to 572°F.

Cooling Time

Since it is necessary to keep the soldered parts from moving until the solder is solid, and since the range of cooling increases as the tin content decreases, it follows that the lower the tin content the longer is the time during which the parts must remain immobilized.

More Solder Necessary

Under some conditions a larger quantity of solder will be required if a solder with a lower tin content is used. There is no economy in the lower percentage of tin if a larger total amount of tin is used.

Silver

2% to 5% of silver and the balance lead. Makes a good solder—but with a higher melting point. Small amounts of silver in a low tin solder also helps, to make a strong solder and a good joint. Even 5% of silver does not make an important difference in the cost.

Bismuth

A small amount of Bismuth will materially reduce the melting point of solder.

Electrical Conductivity

Sometimes alloys are chosen that are required to carry electrical current, and in such cases the electrical conductivity of the various alloys is important. The following are the approximate electrical conductivities of some usual metals based on copper as 100:

85, 5, 5 and 5	16.4
Composition G	13.5
Silicon Bronze	6.5 to 15
Manganese Bronze	24
Malleable Iron	5.4
Cast Iron	2 to 12
Steel	3 to 15
Lead	7.8
Nickel	16
Silver	106.4
Tin	15
Zinc	30
Monel	4

Estimates of the electrical conductivity of alloys based on the known electrical conductivity of the separate con-

stituent metals should never be attempted because there is absolutely no correlation. The above data on electrical conductivity should be considered relative because the conductivity varies with the temperature.

Thermal Conductivity

Based on BTU per hour per square foot per degree Fahrenheit per foot thick, copper is rated 238. and silver 258.

Reducing this to a basis of silver as 100, the following is a guide to the thermal conductivity of various metals:

Copper	92.3
Composition M	19.5
85-5-5 and 5	13.1
Composition G	15
Manganese Bronze	24.1
Si. Bronze	8 to 11.7
Navy Brass	31.3
Aluminum	32.9
Nickel	13.3
Tin	13.5
Lead	7.8
Cast Iron	14
Malleable Iron	15.5
Monel	5.5
Steel	9.7

Estimates of thermal conductivity of alloys based on the known thermal conductivity of the separate constituent metals should never be attempted because there is no correlation. The thermal conductivity of an alloy will always be much lower than the weighted average of the conductivities of the constituent metals.

Brinell Hardness

The Brinell hardness of various compositions is as follows:

Gun Metal Composition "G"	60 - 70
Composition "M"	60 - 72
85-5-5 and 5	52 - 65
Phosphor Bronze	up to 300
Silicon Bronze	100 - 130
Manganese Bronze	150 - 200
Copper-Aluminum	100 - 200

CONCLUSION

It is expected that readers of this report will consider its statements merely as guides and that they will themselves check up on all data given through sources available to them. The comments regarding various bronzes covered in Sections 3 and 4 have been collected from many sources. Every effort has been made to check the accuracy of all statements. Readers of this report, however, should themselves refer to specifications published by recognized authorities. It is not intended in this report to imply that the suggestions contained will apply to every situation. Items judged significant were selected for mention because it seemed that they would probably apply and be worthy of consideration in many different situations.

A NEW APPROACH TO DESIGN ORGANIZATION AND PERSONNEL

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was born September 6, 1911, at Somerville, Mass. He graduated from the U. S. Naval Academy in June 1934, from the Submarine School in June 1937, and from the Post Graduate Course in Design Engineering, USNA, 1943. In 1946 he took the Advanced Management Course at the Harvard Graduate School of Business Administration. He has served in all departments on cruisers, as a Naval Aviation Gunnery Observer and in submarines. He was designated for Engineering Duty only in 1946 and since has served as Deputy Head Electrical Branch in the Bureau of Ships and at the Portsmouth Naval Shipyard where he is now Design Superintendent.

INTRODUCTION

Modern design engineering is largely a group effort; one in which the group is very small compared to the production force. It is a task in which, for example, three hundred engineers evolve material which initiates and directs the labor of six thousand others building a complex warship in a Naval shipyard. In industry, less than five percent of the employees conceive and specify the products fabricated in enormous corporations. All Naval officers are, at some time in their careers, associated with design or design problems; civilians in a technical industry need to be acquainted with that part of their organization which initiates its product.

Since a misdirected effort, a poor design, or a mistake, by so few affects so many, a close look at design organizations and their personnel will enable us to understand and use them better. In a technological nation, whose survival depends upon the excellence of its engineering, the design engineer and his organization are vital cogs. They translate the laws of the scientist into the needs of the operator and consumer. Any deterioration of quality in this field will immediately spread tenfold into poor products, higher costs, unnecessary production effort, and a fatal lag in the output of advanced material. In this article attention will be focused on the basic principles and needs of design organizations, the types of people who work in them, and finally the integration of both into a living organism. Let us look first at the environment in which design engineering on a large scale is done and see if there are certain factors which must be present in all successful organizations.

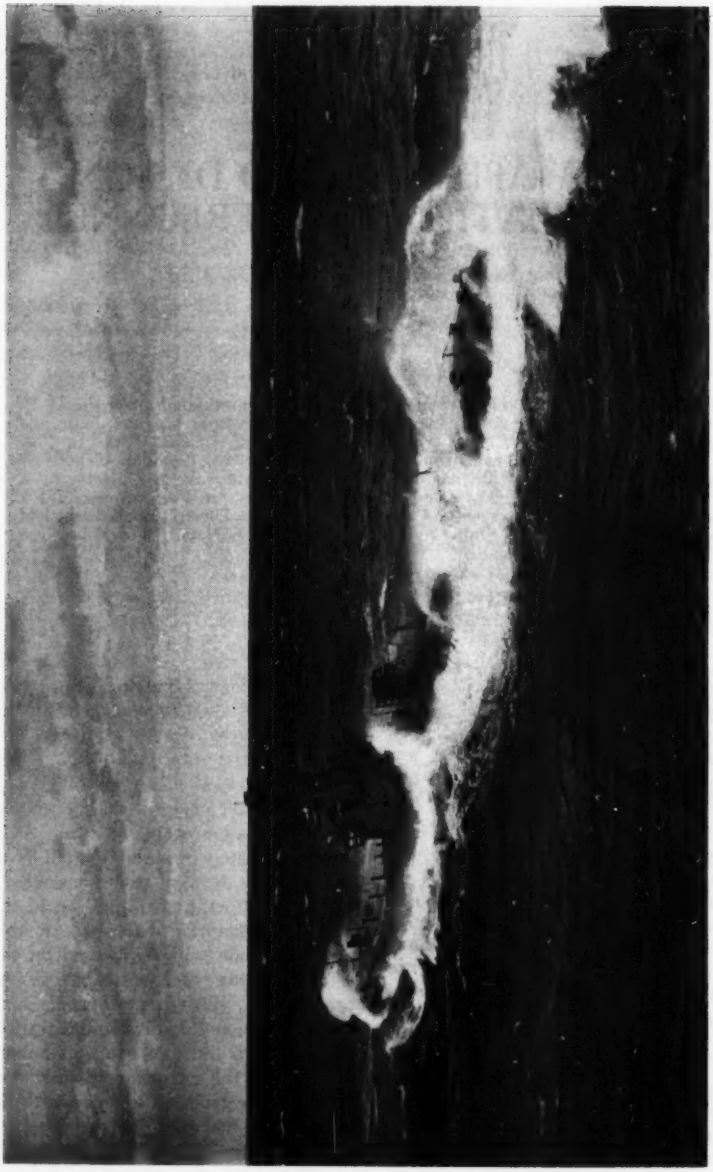


FIGURE 1—"One Hundred Thousand (?) Man Days of Design."

SECTION I—THE DESIGN PROCESS

The Creative Function

The purpose of a design organization is to create a usable product and tell others enough facts so that the product can be manufactured, assembled, tested, operated, and fitted into an arrangement of other products. This simple definition contains many pitfalls to trap the unwary—the most significant words are *create* and *tell*. If a Design organization is divested of any of the above fundamentals its product will suffer. Too often designers are diverted into tasks which others can do equally well—to the detriment of scarce engineering manpower.

Given a complex task, how does a Design Organization create? It is interesting to note that design uses a different approach than research. The designer does not proceed by the scientific method of assembling facts, fitting them to a theory, and, by testing against further facts, deduce a law. The designer and his organization know what the end result should be—it is their task to build this end result from the facts already available to them. The end product is new; the many bases from which it is built are old. This explains why education, experience and varied talents are so necessary to achieve the best designs, particularly if the end result is as complex as a warship or an automobile. Creation, however, is not solely a mechanical assembling of knowledge into a new whole—it involves and must have available to it *imagination*. The organization must have personnel who can visualize an unknown or it will create nothing and merely copy or twist other products into variations of the old.

Assuming a complex design, so as to embrace all factors, the following method of successive approximations seems to fit observation of most design organizations at work. The end product is visualized as a whole, within whatever space, weight, speed relation, and uses have been specified. The system and components which might achieve the specified whole are identified and separate studies are made on each. Such calculations as are needed are used to test feasibility and establish strength of the system and components. The components are then fed back into the approximation of the whole. Their sum total usually does not achieve the desired result and another series of studies is made changing the system or components so that they change in the direction of reducing the error in the desired result. At this point judgment must be used—in many cases the result cannot be achieved and it is more feasible to change the desired whole slightly than to use endless time, money, and effort to attempt a result which is not practical. This is often the case in shipbuilding when a poor compromise has been made in the specifications given the designers regarding speed, displacement, cruising range, and functional components. If the organization assumes a spirit of approximation in working out a design, even of a simple component, it will avoid rigidity, overstressing of one part of the whole, and the fatal error of proceeding firmly along many lines too soon, only to find that one laggard component cannot be fitted into the whole and thus all past effort is useless.

Creative design takes knowledge, imagination, patience, and work—no flashes of intuitive invention will produce a *B-29*!

Putting Ideas on Paper

During the creative process the organization must put its thoughts on paper—first to enable its own approximations to be fitted to that of others and secondly so that the end products can be produced. Many people and some organizations ignore the importance of this phase—the "telling" by means of plans, specifications, instruction pamphlets or work orders.

The organization must, therefore, have skilled personnel who are able to reduce sketches, calculations and ideas to clear readable blueprints. Good drafting is not just drawing. It requires engineering skill, knowledge of materials, and a three-dimensional mind. Later when we discuss personnel this problem of drafting will assume the proportion of a large problem, as the area between drafting and engineering is an overlapping one. The clear presentations on paper of creative design is a most important function. Dimensioning, clearances, tolerances, threading, welding, filleting—all must be thought out, and carefully drawn, or the product will not achieve the expected results. Insofar as possible, drafting standards should be promulgated so that all plans will be interpreted the same way.

Plans are usually not enough to complete a design, however. Usually specifications for materials and operating instructions are necessary. Specification writing is an art in itself. It requires a command of good precise English, the ability to state *exactly* what is wanted and a keen appreciation of the balance between too much and too little. The former results in an impossible specification; the latter, in a shoddy product. Operating instructions, too, are a specialized field—one in which the writer must take technical terminology and translate it into clear everyday English.

Testing

Finally, the organization must have a means of testing its output as the design proceeds. No design, if it is a new creation, will be correct and perfect the first time. Testing must determine the feasibility of each component and system as the design proceeds. Here we enter an organization area in which there is much controversy. Some organizations have testing personnel attached to, and integral with, the design organization. In others a design is tested by the production department. Some have testing divisions which are a part of inspection and separate from design or production. The Navy is itself divided on this problem and examples of all three are found in industry. The best compromise is possibly a small Design testing group which can set up engineering tests for those components or systems which are felt to be new or controversial, while the whole design is tested outside the organization, and fresh or consumer viewpoints are obtained. In this case free communications are a must and the Design organization should see the manufacture and operation of its product, so that facts may be funneled back for change in design. Too often Design changes are based on uninformed comments or on the opinions of a narrow segment of the products' potential users. Adequate testing will prevent this.

SECTION II—THE DESIGN ORGANIZATION

Formal Structure

There are many forms of Design organization existing. Some combine many of the above functions in one group—all of the successful ones however have the qualities of "create" and "tell" inherent in them. Information is freely interchanged and there is unified direction. An example of a typical Design Organization is given as Figure 2—the standard organization for the Design Division of a Naval Shipyard. Its Arrangement Branch initiates the design approximation by first blocking out the gross space and operating location of major components from Bureau specifications and plans. The technical sections decide what type of components will be used and proceed to design them while its Scientific Section checks the over-all weight balance, strength, and hydrodynamic features. The test section sets up tests of systems or components as requested by the technical sections. The Liaison group schedules, gives management key data for control, and keeps design informed of Production problems, while the Service Section provides blueprinting, filing, messenger and other services to all.

In a smooth organization free horizontal flow of information between branches is affected while the Staff Officers and senior supervisors provide the guidance necessary to keep the parts working at an even pace to achieve a balanced design. When looking at Figure 2, it is interesting to ask, "At what level and where does the creative process take place?" The bare bones of an organization do not tell—we must examine the people available. This will be done under Section III.

The Design Organization is never wholly a command structure. This point is most important to military personnel whose previous training has been with a military structure where organization must be based on command responsibility and a vertical flow of information.

Interchanging Ideas and Information

Responsibility and decision making must, as in a command or staff organization, be carefully delineated. However, a much freer flow of knowledge between Branches is necessary than the information which flows between the engineering and deck departments of a ship or the Sales and Manufacturing Divisions of industry. Going back to the method of successive approximations given above, it should be readily apparent that it would be impossible for the thousands of studies and ideas needed for a complex design to travel up the line to the head, down to another specialized group, and back through the same channels. This flow can be accomplished by designating a "lead" branch or section responsible for a system or component even though other types of engineering than its specialty are involved in its design. The other branches which have a service or minor part in the particular design must keep the lead code informed of any decisions affecting its cognizant items. Higher echelons than the lead Branch may then have an over-all study or plan of a system without having to check many groups for the information in order to study the integration of one system with another.

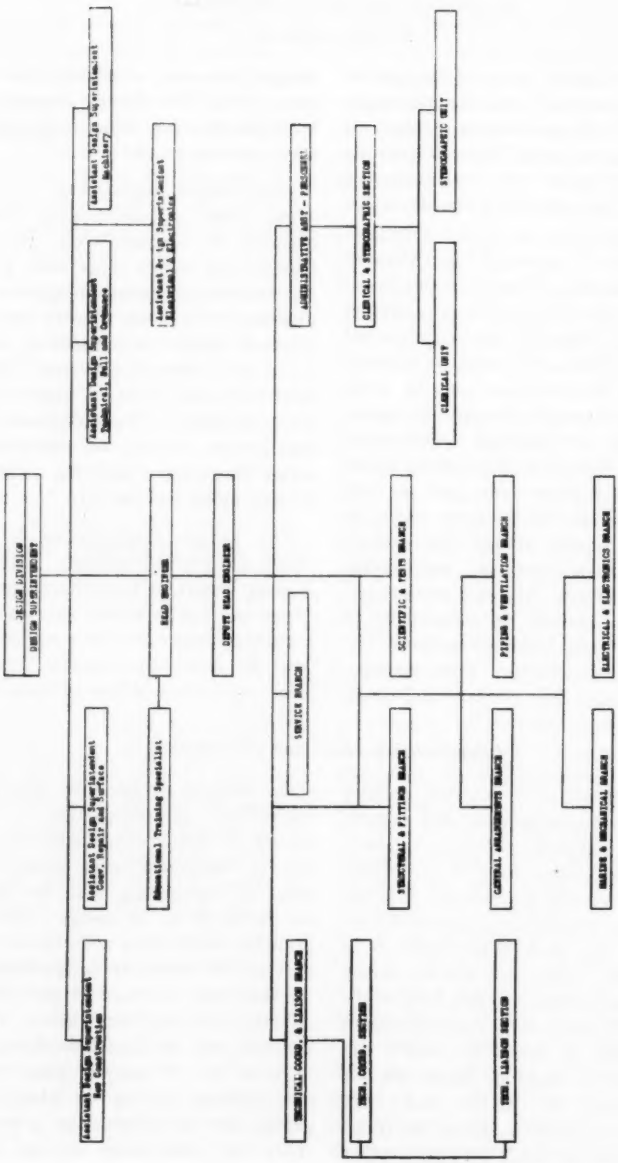


FIGURE 2—Chart Showing the Design Division of a Naval Shipyard.

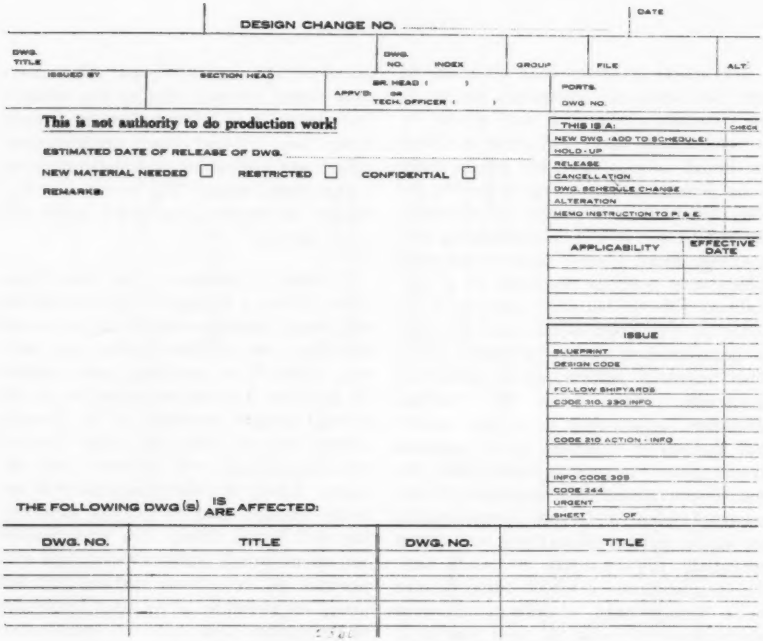
Project Groups

The above assumes that all tasks fit into the functional groupings set up in the structure chart. This may not be so, as occasionally tasks will arise in which the result is in a field for which there is no definite group set up to handle the job. Such a task must be carefully studied to see if it is a continuing one resulting from a new engineering field which may continue to grow, or a specialized job to be done just once or twice. If the latter is the case the task is best handled on a "project" basis. This involves setting up a group of specialists taken from the various Branches, appointing a project leader and having it function as a separate group under the management until the task is completed. An example of this occurred when a Design Organization set up to design submarines was given operating requirements to work out, the specifications of which could be met by a surface ship, a submarine, or a combination of both. If sent out as a routine task to the organization with its concentration on submarine design, the result would have undoubtedly overstressed the submarine aspects. By picking out personnel who had previous experience in both types, setting up a separate group, and making available outside consultants who had experience in the phases the group lacked, a balanced study was achieved. The above may sound obvious, but to the writer's knowledge it is seldom done and many tasks are ground into organizations which are not set up to handle them on a functional basis. The result is a bad design no matter how much good will goes into it.

A note of caution is due here, however. When a Design Organization has too many projects outside of its normal structure, one of two factors are present: either it is accepting tasks outside its field; or it is not organized to do the normal designs expected of it. Drastic action must be taken in either case or the organization will produce poor designs. Lines of communication will become blocked, and its normal functioning will break down. An organization set up to handle submarine design may produce an occasional surface ship but never an airplane. It is most important to recognize this fact in commercial industry during this period of cold war. An organization set up to produce commercial designs in peacetime can handle some wartime designs but when military design becomes the normal job, the organization must be shifted to handle its new tasks. Careful executive planning is necessary to accomplish a gradual shift.

Changes

Another organizational tool which must be present and which does not show on a chart is a fast, direct, standardized method of handling changes. We stated before that changes are inherent in design—they must and will be made. It therefore does no good to berate the changes or order a freeze of design too early. The problem should be handled directly by management and a system set up so that changes are disseminated to all who are involved (production included) and these which affect the whole or will retard schedules be brought to top management's attention. Figure 3 shows a design change sheet which will accomplish this purpose. This sheet combines all that is needed and gives management a control which can be used to stop unnecessary changes and to check the effect on erection schedules.


DESIGN CHANGE NO. ..... DATE

DWG. TITLE | DWG. NO. INDEX | GROUP | FILE | ALT.

ISSUED BY | SECTION HEAD | BR. HEAD ( ) APPV'D: OR TECH. OFFICER ( ) | PORTS. DWG. NO.

This is not authority to do production work!

ESTIMATED DATE OF RELEASE OF DWG. .....

NEW MATERIAL NEEDED ☐ RESTRICTED ☐ CONFIDENTIAL ☐

REMARKS:

THIS IS A:	CHECK
NEW DWG. (ADD TO SCHEDULE)	
HOLD-UP	
RELEASE	
CANCELLATION	
DWG. SCHEDULE CHANGE	
ALTERATION	
MEMO INSTRUCTION TO P. & E.	

APPLICABILITY	EFFECTIVE DATE

ISSUE	
BLUEPRINT	
DESIGN CODE	
FOLLOW SHIPYARDS	
CODE 310, 230 INFO	
CODE 210 ACTION - INFO	
INFO CODE 205	
CODE 244	
URGENT	
SHEET OF	

THE FOLLOWING DWG (S) IS / ARE AFFECTED:

DWG. NO.	TITLE	DWG. NO.	TITLE

FIGURE 3—A New Form of Design Change Sheet.

Scheduling

One frequently hears the assertion, "Brains cannot be scheduled." The inference is that a design organization cannot schedule its output. This is an over-simplified generalization which if followed will lead to chaos in a design organization. It is true to state that a human brain cannot be forced to create an entirely new concept in a given space of time. The statements might be applied to research but, in general, they are not true for a Design Organization. Design can, and should, be scheduled—since parallel production and purchasing groups *must* know when plans and purchase specifications will be available for an orderly erection schedule. Design must, for its own sake, schedule interdependent plans. This phase of Design is a complex and difficult process, involving all levels of the organization. Scheduling *things* is complex—books have been written on the subject—scheduling an output of *ideas* is infinitely more so. This article is primarily concerned with over-all organization and personnel so that scheduling can be treated only lightly. Several precepts can be given, however.

First the task must be broken down into a list of plans and instructions needed by the producer to make the article and components. Usually this list can be further broken down into what is new and what is old. Plans which depend on known concepts or a slight modification can, with experience, be scheduled directly. These plans em-

bodying a new system or component are the keys to the design schedule and must be carefully scrutinized to determine whether development or research is necessary. If so, the time factor depends on judgment, but an estimate must be made. The estimate should be broken down into phases:—determination of the best field of engineering, the time to first model. testing of model, determination of changes and time for final detailing.

The entire plan list must then be dovetailed together such that a *sequence* of plans is determined and no plan is listed for completion ahead of another on which the first plan depends. Production should now be consulted since an erection or manufacturing sequence also determines the order in which plans are needed. These two time demands are often not compatible and must be reconciled. An example of incompatibility often found in shipbuilding is that of the timing of foundation plans. The builder wants them early in construction since they attach to the shell, are bulky and form interferences for wiring and piping. The designer cannot draw a foundation plan until he has determined the size, shape, and weight of a component, even though he knows generally what the article will be.

At this preliminary schedule stage design manpower can be allocated and an estimated curve of manpower against time drawn. Dates are set for each plan completion and a plan schedule for the entire task printed. It usually cannot contain *all* plans which will be necessary, as the designer does not know at this time all plans which will develop—80%-90% is a good figure to shoot for. The plan schedule should now be compared with the erection schedule to see that it fits. Control curves should be drawn so that the Design management can ascertain the following at any time:

a. How manpower being used compares with estimates

b. How the output of plans is following the plan schedule

c. How specifications written compare with the lead time necessary for purchase of materials. Examples of each of these three curves are given as Figures 4, 5, and 6. There must be a group to do this over-all scheduling and control.

The bare bones of an Organization Chart and a description of its formal functioning do not describe the design process adequately. *People* design and therefore we turn to the personnel of a Design organization—their types and how they work.

SECTION III—DESIGN PERSONNEL

The Engineer

The generic term "engineer" is used so broadly in our language that it has lost its original precise meaning. It can now be used only in the broad sense that "Christian" is used to describe people of a general religious belief. Nevertheless, we still use the word "engineer" in its broad sense to attempt to describe all sorts of specific work. This leads to confusion when describing talents necessary to perform a design job. Gobbledygook creeps into classification procedure where names such as "scientist," "sub-professional," "engineering aide," "draftsman," etc., are used by the Civil Service in an attempt to graduate the degrees of design skill needed for a job. I venture to say that an honest appraisal of any Design Organization would show people titled with some of the above names doing wholly or partially the work nominally assigned to other classifications because of this confusion. This comes from another dangerous notion fixed in many non-technical minds who control the fu-

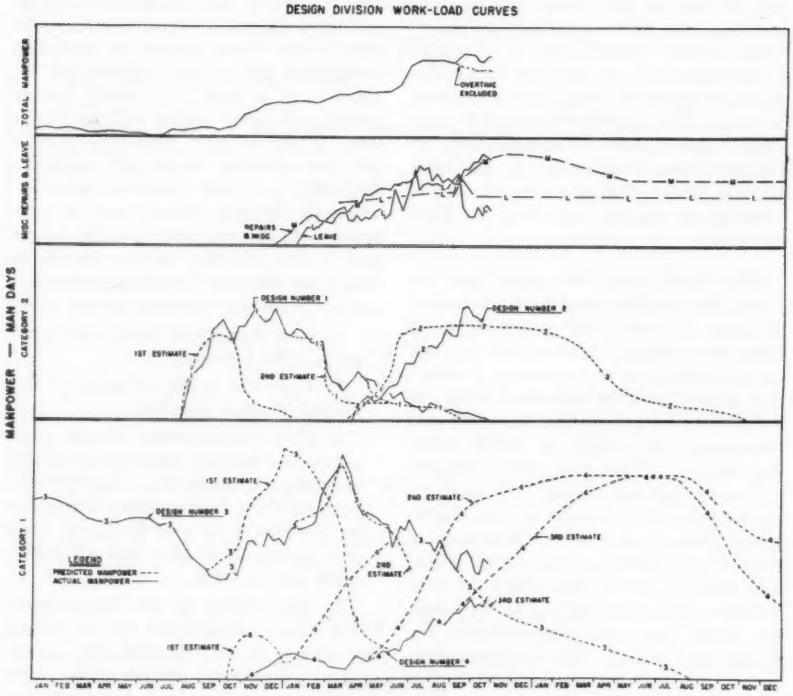


FIGURE 4—Manpower Utilization Chart.

ture of Design and its people. This fallacious concept of Design (another too broad term) seems to consider that an "engineer" is one who sits at a desk, produces a design full blown out of his trained mind, reduces it to a set of calculations and then tosses it to a draftsman who thereupon draws a plan. Nothing could be further from the actuality of design practice. Some designs are sketched or drawn first, then calculated—some are designed completely on a drafting board and need no mathematics and some combine both—the methods are infinite. All methods are necessary. All are used. No design organization could survive if it was composed of engineers who did no drawing, and draftsmen who did no engineering. In order to avoid confusion in semantics this article will use terms which symbolize the present functions of designers rather than their popular titles. Depending on the size of the organization, some types will be combined in one person, but in any organization personnel must be available to do the types of work described, and the organization must have them placed at the right levels to work effectively.

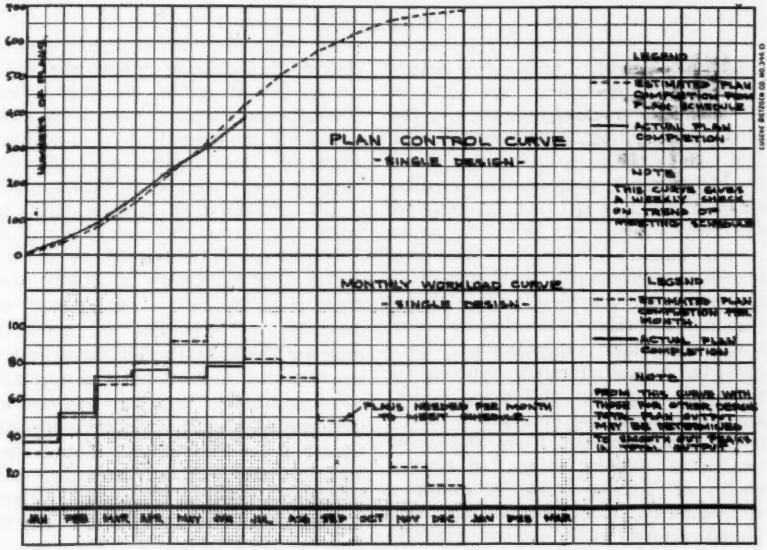

PLAN CONTROL CURVE
-SINGLE DESIGN-
LEGEND
ESTIMATED PLAN COMPLETION FROM PLANT SCHEDULE
ACTUAL PLAN COMPLETION
NOTE
THIS CURVE GIVES A WEEKLY CHECK ON TREND OF MEETING SCHEDULE
MONTHLY WORKLOAD CURVE
-SINGLE DESIGN-
LEGEND
ESTIMATED PLAN COMPLETION FOR MONTH
ACTUAL PLAN COMPLETION
NOTE
PLANS NEEDED PER MONTH TO MEET SCHEDULE
JAN FEB MAR APR MAY JUN JUL AUG SEP OCT NOV DEC JAN FEB MAR

FIGURE 5—Plan Control Curve for One Design.

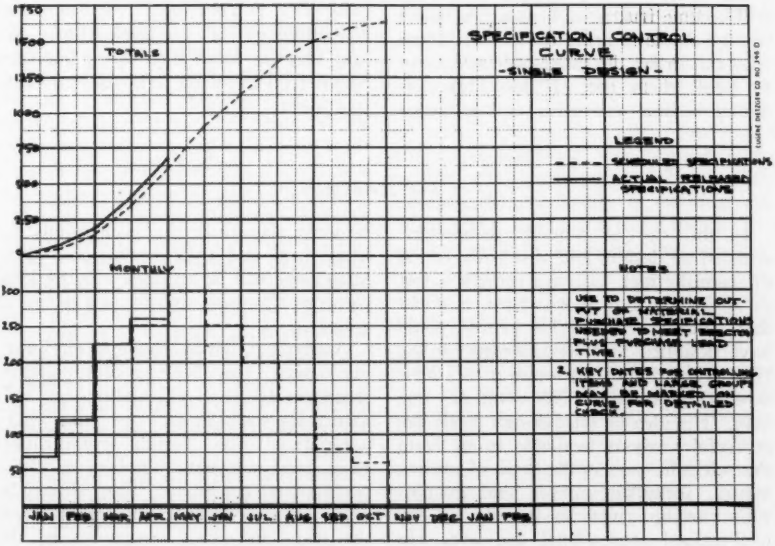

SPECIFICATION CONTROL CURVE
-SINGLE DESIGN-
TOTALS
MONTHLY
LEGEND
SCHEDULED SPECIFICATIONS
ACTUAL RELEASED SPECIFICATIONS
NOTES
USE TO DETERMINE OUTPUT OF MATERIAL PURCHASE SPECIFICATIONS NEEDED TO MEET REQUIRED PLUS PURCHASING LEAD TIME.
2. KEY DATES FOR CONTROLLING ITEMS AND LARGE GROUPS MAY BE MARKED ON CURVE FOR DETAILED CHECK.
JAN FEB MAR APR MAY JUN JUL AUG SEP OCT NOV DEC JAN FEB

FIGURE 6—Specification Control Curve for One Design.

The Idea Man

The more creative minds in a design organization, the better; however, not all should be or can be creative in the sense of imagining a new concept. The idea man is the fellow who says "Let's try it this way." Then he may verbally, or by a sketch, rough out his proposal for others to work out, or he may draw it out himself. It is just becoming known to psychologists that truly creative minds are very scarce. It is therefore not surprising that they are also scarce in design organizations. Fortunately, a well trained, experienced technical man can, and does, evolve answers to technical problems by combining his knowledge from books and experience, without using that elusive quality of creation which, for example, an artist has. Truly creative idea men should be found at all levels of a design organization. The idea man should be concentrated either at or just below the unit supervisor or project leader level. If too low in the organization, he will become frustrated because he hasn't enough status to bring his ideas to those who can order the manpower necessary to work them out. If too high, his duties become executive and administrative to the point where the daily press of work stifles creative ability. The idea man is there—the trick is to find him and place him in a position where his ideas can be seen and used. He may have to be given a fictitious title in the organization but he must be found and used if the designs are to be progressive. Paradoxically, a college degree does not endow its holder with creative ability, but a degree may be necessary to work out the germ of someone else's idea! Let me repeat—the idea man must be spread throughout an organization; he must be given an opportunity, and he cannot be requisitioned by merely specifying engineering qualifications. You may find him in a top management job burdened by paperwork and personnel problems. Or he may be at the bottom tracing someone else's design, wondering why it wasn't done another, and much better, way. When you find him—*do* something about him—he's a rare bird! Edison was contemptuous of mathematicians and said that he could hire them, but they couldn't do his work. He had the creative mind!

The Analyst

In this day of complex technology, speeded-up design, and vast background of scientific fact, Edison's remark is not as all-inclusive as it was then. I doubt that Edison, with all his genius, could have designed a turbo-generator in the size, weight, and horsepower of those now being produced. Nor could any one man. The idea or trial design needs the powerful analysis available to us from the scientific past in the form of mathematics, formulae, curves and graphs. This can be given to it only by the well-trained professional engineer. He must not follow handbook rules solely; he must also know the assumptions behind them and their limitations. This knowledge can be attained only by long rigorous training—usually in a technical college.

Analysis is the backbone of engineering design: the checking of a complex structure by such calculations as will assure its strength and operating characteristics. The design organization must have adequate numbers of these professional engineers. They should be experienced personnel who are aware of the ways in which operations of the product seemingly invalidate safety fac-

tors and formulae. They should be continually replenished from the bottom by young graduate engineers whose minds are freshly sharpened by college science and who are acquainted with the powerful new tools of mathematics that are continually being evolved. No one can take the place of these men—their importance is attested by the strenuous efforts being made to recruit them from colleges and the national effort to channel more young men into technical colleges. No derogatory implication is meant by the heading *The Analyst.* It should rather read: Those who have the training, inclination, and ability to analyze mathematically. There are, of course, many idea men among them—the point is that all idea men are not necessarily mathematical engineers, nor are mathematical engineers necessarily full of new or novel ideas.

The Draftsman

The success that a playwright brings to an author whose works are converted to the stage is the type of success that draftsmen bring to those who have conceived, computed or sketched a design. The draftsman in the pure sense lays out the lines, evolves the inter-relationship of parts, sets the fits and tolerances, draws the bill of materials and puts on paper the myriad details necessary to make a dream an actuality.

As we mentioned before, his status becomes controversial as his experience and ability grow—it is not unusual to see design organizations still headed by a man titled Chief Draftsman. The chances are that this man is not a draftsman in the popular sense of the word, but rather a highly skilled engineer in his particular field. It was this fact which prompted the Bureau of Ships of the Navy Department about 1947 to change shipyard design organizations into engineering structures. Since that change, the organizations have gone through many stresses in differentiating between draftsmen and engineers. In the middle ranks the experienced draftsman is a man who usually has wide experience and knowledge in one engineering field. To do his job, he must know shop practice, production methods, and how an article is fabricated so that his layout will be understandable to shop personnel. He knows drafting standards, descriptive geometry, handbook formulae in his field, and is a skilled drawer. The draftsman is the indispensable partner of the degree engineer. If there is a barrier between them, the end product will suffer.

The draftsman is best recruited from shop personnel and requires special training for his job. Once he has attained drawing skill and studies the theory of the science in which he is working, he tends to grow closer to the professional engineer in his ability to create designs in his particular field. At this point his talents are often much more valuable than the younger design engineer who has had no working or shop practice. This causes confusion as to who *is* the engineer and if pay scales are not comparable, will cause friction within the organization. To further compound the difficulty of defining and identifying the draftsman's work, the young engineer gains his best training and practice by either doing drafting work, or working with an experienced draftsman to gain knowledge of local shop processes and experience which no school can teach him. The whole subject of the inter-relationship between the draftsman and engineer has not been given the study it deserves, and clarification is needed by the Civil Service Commission, the Department of the Navy, and classification experts in Industry.

The Supervisor and the Higher Degree Problem

Volumes have been written about the Supervisor, his characteristics and his duties. The necessary qualities of leadership, ability to plan, etc., are just as needed in a design supervisor as in any other type. His growth from the lowest grade supervisor to the top, however, forms quite a different pattern and has received little attention.

It is sometimes thought that the more degrees in engineering a man can attain, the greater should be his ability to design. This is true in a particular field and if the man be considered alone. In an organization, however, if there is no ladder to travel up but the supervisory one, he may become of decreasing value! This is because the amount of detailed technical ability needed for degrees becomes subordinate to the amount of executive ability demanded as one manages a larger group whose product includes more and more fields of engineering. The mastery of one subject to the Doctorate level is *not* consonant with the mastery of executive ability. When there is one pay scale, based on supervisory advancement, as there is in most Service design organizations, the two roads become confused and the organization suffers whichever one it takes. A design organization loaded with solely technical-minded specialists at the top is as apt to falter as one with great executive ability but no firm foundation of professional knowledge inherent in its management.

There seem to be two solutions to this problem. There is a level in any design organization where technical proficiency is the most needed—where decisive individual engineering is done. This is usually just below or at the unit supervisor or project leader. Here a master's degree, long experience in drafting, or unusual ability to create are most effective—supervisory ability is subordinate. From this point on, the need for executive ability grows in proportion and training in this field should be initiated. Promotion policy should be made on a weighted scale in which executive development is mandatory, although greater technical excellence is encouraged. Somehow the super-technical man must, however, be rewarded with increased pay even though he is not a leader, and has no administrative ability or leaning. After all, Steinmetz had neither, but who can measure his value to the GE Company and his country? Therefore, the ideal design organization should have a structure which demands executive ability in its supervisory ladder and also has a place with adequate pay for its personnel who have advanced technical or scientific minds. How many times we see brilliant technical minds struggling with the personnel and administrative paper work which an executive could wade through with ease, while a potential leader is making errors with a slide rule which he hates! Most men are ambitious, all desire advancement in pay —some channel must be provided for both types.

Other Types of Personnel

There are many other types of designers found in design organizations. There is the test engineer—a link between design and operations whom we mentioned before. The arrangement man —he of the three-dimensional mind who fits the pieces of the puzzle together to form an operational whole. The checker —half way between a supervisor and a draftsman or engineer, who is a com-

bination Sherlock Holmes in finding errors and a Mr. Chips in getting these errors to resolve themselves without destroying initiative. We mentioned before the technical writer and the specifications writer, both needing the gift of precise English. All are necessary and all must find a rewarding place in a good design organization.

The Executive Engineer

One type we have not mentioned—one which is seldom defined but which is always outstanding in an organization—the Executive Engineer. Perhaps a typical advertisement such as you might find in the *New York Times* might define him. "Wanted:—Engineer with full professional background, preferably ME, with knowledge of production processes and ability to handle personnel, to manage growing enterprise in a new field. Unlimited opportunity." Why do they advertise? The answer is simple—there is no one in the present organization who can fill the bill! This is not an indictment of such organizations—it is rather to point a crying need in design organizations of all sorts in our country—the need for engineering *managers*. In this age of specialization and competition, the technical man has so much study to master his field, the administrator so much to do that he has no time for technical broadening and the combination is rarely found. Seldom is it so trained. There is not space to describe the dire necessity which initiated such programs as the Harvard Business School Advanced Management Course for top executives during World War Two, nor the many problems when an outstanding man in a technical field was promoted to a top job covering many fields and failed miserably. Design organizations need executive engineers—they must have them or they, too, fail. Most Naval Officers in design organizations are really executive engineers rather than design engineers, despite their designation. Top supervisors should be in this category.

It is the executive engineer who keeps all branches or units heading towards the same goal at the same pace, who fills in the gaps of organizational knowledge by drawing on consultants or recruiting, and who gives his organization the stimulus of enthusiastic leadership. The executive engineer is not necessarily, the top man—he should be found in staff positions and at various levels down to the Section supervisors. It would be easy to say that all managerial personnel should be executive engineers, but this is impossible—there are not enough to go around. They should be placed where their influence is the most needed, and above all, trained for higher positions along executive lines, continually broadening their technical, administrative, and supervisory abilities.

SECTION IV—CONCLUSION

Much has been said of late in the Press and official circles of the shortage of engineering manpower and its alarming implication for our nation. Articles such as "Operation Design" have shown the efforts of organizations such as the Bureau of Ships to train, nourish and enhance this technical manpower. There is a need, however, to study the designer with his organization. Design must be studied the way production is studied, so that each type of person is fitted into a position which will use his talents to best advantage. Design organizations must be used to *create*, not to act as production instruc-

tors or trouble shooters. They must be organized so that the formal structure contains the elements of supervision and responsibility but does not lack the flexibility and interchange of ideas horizontally so necessary as design and technology change.

Classification must be the servant rather than the master of personnel placement. To do this classifiers need either to have technical knowledge themselves or to rely on technical management to set up job descriptions which fit the actual work done by designers rather than popular generalizations or outmoded classification standards.

Those who use the products of Design organizations help themselves when they understand the process of design and the people who do it. They will better understand why changes are not mistakes, but rather an integral part of the growth of any design. They will better know where and to whom to address their questions and problems. Finally, the training of Design personnel must be tackled in *all* of its aspects—in the wholly technical area, the drafting area, the supervisory area, and the executive area. It is hoped that this article will stimulate broad thinking on the whole subject of the designer and his organization and will aid in the conservation and concentration of a precious portion of our human resources.

MILITARY SYNCHRO-CONTROLLED SERVO-SYSTEM DESIGN NOTES

TED POWELL

THE AUTHOR

Born in Albany, N. Y. on October 20, 1914. Attended Newtown High in Elmhurst, L. I., N. Y. from 1928-32, and studied electrical engineering evenings at C.C.N.Y. and Pratt Institute in N. Y. C. and Brooklyn from 1932-5. Was employed at the N. Y. Naval Shipyard in Brooklyn from 1937-46 where he helped organize, train and operate the civilian Ship Repair Force Electrical Test crews of the Electric Shop and prepared field test instruction manuals during the recent war. Was with the Oscilloscope Engineering Department of the Dumont Labs at Passaic and Clifton, N. J. from 1946-7, the Engineering Department of the Amplifier Corporation of America, in 1947, and the Engineering Department of the Hazeltine Electronics Corp. from 1947-51. Is presently with the affiliated Advanced Research & Development Section of the Hazeltine Corporation of Little Neck. Has done intermittent technical article writing on various electronics, electrical and military subjects since 1937.

This non-technical paper discusses some of the field test and maintenance problems encountered with World War II military servo drives of the closed-loop continuous-control type. Such troubles were not always readily foreseen in the engineering lab and sometimes cropped up only after long and severe service under diverse field and combat operational conditions. This was often aggravated somewhat where less experienced military personnel was available for necessary trouble-shooting and maintenance routines. The rather obvious thought here is that perhaps ordnance and electronics engineers might predesign their equipment so as to provide a certain amount of field-operational "anti-snafu" factors into future military designs.

THE SYNCHRO SIGNAL CIRCUITS

Some manufacturers of wartime indication and control synchros (selsyns) did not standardize their designs completely and did not always fully meet the military specification which requires that all synchro units should be so wired internally that a standard test synchro connected to any terminal board anywhere in any synchro system, will indicate increasing readings when the transmitting equipment is actuated in the increasing-angle direction, and read zero when the transmitter is set at zero. This lack of complete standardization on dial engraving rotation and synchro winding electrical rotation, usually resulted

in reversed reading and reversed indicator rotation troubles even though the systems were wired correctly according to the installation blueprints. This sometimes resulted in rounds of dial-shifting and synchro wiring reversals during installation test routines, especially where various synchro systems had to be cross-tied with existing systems. The reversed-signal and reversed-rotation troubles were caused by some half-dozen different conditions.

1)—Some synchro units' dials were calibrated in a clockwise direction, and others on identical units of different manufacture in the same system were calibrated in a counterclockwise direction. Hence although all synchros rotated in the same direction and zeroed in with their transmitter properly, some of the dials indicated reversed readings. The S1-S3 or R1-R3 synchro leads of the offending instruments had to be reversed at the instrument wiring side of the terminal blocks to reverse the rotor rotation to align up the system, and everything would work out fine. However, if the synchro leads were reversed in the wrong section of the system, trouble would break out pronto when the system was cross-tied in with an existing system.

2)—Some synchro units in a system used rotating pointers with fixed indication dials, while similar units in the same system used rotating dials with fixed pointers. Even though both types of synchro units were wired up internally in the correct fashion, both types of dials were engraved in the same direction, and both sets of synchro rotors rotated in the same direction, the two sets of indicators would read in reversed directions with respect to each other. This resulted from the fact that the relative directions of rotation between the two types of dials and their pointers were in opposite directions. Again, if care was not exercised in reversing the synchro leads at the proper instruments, trouble would be encountered when cross-tying in with other systems.

3)—Military A.C. 5-wire synchro leads are usually tagged R1-R2 and S1-S2-S3 for the 2-pole rotor and the Y-wound stator leads. The specs require that a synchro unit read correctly against a standard test synchro, both as to the electrical zero position and the direction of increasing readings (or direction of rotation in the case of a simple synchro indicator). The mere fact that a synchro's stator leads are marked S1-S2-S3, however, does not indicate what the actual electrical rotation of the winding is. Hence it is possible for a synchro indicator unit to have both mechanical and electrical reversals internally which cancel each other out and permit the complete unit to meet the standard motor test. Hence some makes of synchros had Y stators whose electrical rotation was opposite to that of most other makes of synchros. When such a synchro was removed for repairs and later re-installed into another unit, reversal troubles cropped up and the S1-S3 or R1-R3 leads had to be reversed at the instrument side of the terminal block. A similar trouble existed in the case of 3-phase motors and generators where the mechanical shaft rotation and the winding electrical rotation were not always one and the same thing. Hence the factory production test sheet should specify that both the naked synchro and the complete synchro unit must both meet the standard synchro motor test. This was a troublesome point which was difficult to get across even to ordnance design engineers.

4)—Standard military (Fire Control) synchros use a 2-pole rotor field (R1-R2) and a Y signal stator (S1-S2-S3), while most marine type (Interior Communication) synchros use the reverse combination—a Y signal rotor (R1-R2-R3) and a 2-pole stator (S1-S2). Even though both the electrical phasing and

rotation of the two sets of windings were the same, the two types of synchros' rotors would rotate in the opposite directions. This resulted from the fact that the relative directions of rotation between rotor and stator were in the opposite directions, much as in the case of the rotating-pointer-rotating-dial reversal problem. Hence when navigational I.C. systems were cross-tied in with F.C. gunnery systems, reversals troubles developed. Again, if care was not taken to reverse the correct (1-3) Y-circuit leads, trouble would develop when cross-tying in with other systems.

5)—Some F.C. indicators use concentric duplex and triplex ring-dials and gear-trains for various match-the-pointer and relative-true bearing indication purposes. This brought up confusing reversed-rotation and reversed-reading troubles if the instrument wiring was not properly reversed at the instrument side of the terminal blocks at the factory to meet the standard motor test.

6)—Some multiple-unit equipment such as radar and sonar gear, contained synchros which had their instrument wiring reversed to correct for various gearing reversals and dial calibration reversals. Sometimes the wiring reversals were made at the interconnecting cable wiring side of the terminal blocks instead of the synchro wiring side. When standard test synchro motor checks had to be carried out on such systems, or additional equipment cross-tied or added to it, reversals problems developed. Both design engineers and factory production men must be continually impressed with the fact that all wiring reversals should be made at the synchro wiring side of the terminal blocks to meet the standard motor test, so as to prevent field installation and test confusion later on.

All in all, non-standardization of synchro units during the recent war turned out to be something of an ulcer-generator and this appears to be a fruitful field for military standards committee activity.

Instances of puzzling loss of synchronism and erratic behavior which were usually promptly blamed upon faulty installation wiring by company field engineers, were sometimes tracked down to slipping set-screws on shafts, couplings, dial hubs and gears at synchro assemblies inside of ordnance and radar gear. Apparently, the use of aluminum alloys for dial hubs, gears, and linkages tended to aggravate this trouble because of the creep tendencies of aluminum metals under pressure and heat, and their relatively high thermal coefficients of expansion. Stainless steels appear to be more reliable here, even though heavier, more costly and harder to machine. Gun-fire shock, vehicle propulsion vibration, synchro A.C. vibration and ambient temperature cycling may also have contributed to this trouble. Twin set-screws, shaft flats, shaft keys and set-screw holes in shafts seem to be necessary as an insurance against such operational troubles.

Conical set-screws for synchro dial hubs caused some maintenance troubles since they bit into synchro rotor shafts too deeply and burred them up so much that dial hubs were difficult to drive off and on the shafts. Many dial hub assemblies were conical press fits which were difficult to pry loose. As a result, shafts, hubs and shaft bearings were damaged out in the field when hubs were driven off by various brute-force methods. Some sort of a simple hub-puller should be included with military synchro equipment maintenance gear.

Installation of large-sized dials on modified ordnance units often resulted in synchro hunt and motorizing instability problems since the large dials' excessive mass and torque moment overrode the stabilizing action of the synchro rotor-damper mechanism and caused the synchro rotors to overshoot badly or run away. If the oversized dial was

unbalanced, it would also increase the bracket error on two sides of the dial since its gravity pull would tend to pull the rotor out of synchronism slightly.

Mechanical linkages and gear trains between synchros and their units, resulted in some servo jitter and sync error at static synchronism, and some indicator motor hunt tendencies. Spring pre-loading of gears and linkages, and slotted mounting holes appear to be worthwhile in the case of precision F.C. equipment in the long run, in spite of initially greater cost.

In cross-tying F.C. gunnery and I.C. navigational systems, the precision F.C. synchro bracket error was increased by the marine type I.C. synchros whose bracket error was somewhat greater. The greater winding and core unbalance effects of the I.C. synchros' Y rotors tended to unbalance the cross-tied synchro systems' Y signal circuits slightly to increase over-all system error. This point bears some thought where precision gunnery circuits are involved and indicates the possible need for more accurate navigational synchros in some cases.

Some servos employing synchros as the control elements, also fed remote-indication indicator motors off the control-synchro lines. Any mechanical or electrical troubles (or battle damage) in the remote indicators or their wiring, would derange the control-synchro system and foul up the vital drives. The cost of extra synchro generators for separately feeding the remote indicators may be justified in the case of vital gunnery, radar, sonar and navigational systems.

Special "H" type hi-RPM synchros made their appearance during the war for continuous-rotation duty applications. They sometimes tended to develop brush and slip-ring arcing, burning and over-heating troubles. Apparently, their only modification was a more massive rotor shaft bearing. Some investigation into the need for a heavier slip-ring, brushes and brush springs, more accurately machined slip-rings, and power-factor-correcting condensers across the rotor and stators to eliminate reactive currents, might be indicated for these "H" type synchros.

Most of the radar, sonar and navigational synchro systems used no trouble-light circuits (which consisted of current transformers and their indicator lights in series with the synchro Y circuit lines). A synchro may have its rotor circuit open and it will still follow its transmitter either correctly or 180 out as a sort of a hysteresis type inductor synchro. Similarly, it may have its two rotor leads shorted (but not across the rotor supply lines) and its rotor will snap 90 or 270 out and follow the transmitter as a sort of a wound-rotor induction synchro. However, in both cases, the fouled-up synchro will have weak torque and will pull its excitation power from the transmitter synchro by means of a double-transformer action via the two synchros' Y windings and the two units will overheat and may burn out in time. Hence costly synchros were damaged sometimes when the operational troubles were not noticed soon enough. The use of trouble-light circuits in such systems may be worthwhile in view of the high cost of precision synchros.

Many gunnery systems' synchro circuits were modified at some time or another to take additional control or differential synchros. Since these synchros have no field winding, they take their power excitation from the generator synchro, with the aid of p.f.-correcting delta banks of condensers across the Y signal lines. Some confusion and fussing with synchros systems resulted since trouble lights would come on and the alteration prints would give no info on re-setting the trouble-light transformer sensitivity taps to allow for the extra required excitation currents.

Some indicator unit synchro motors had no pre-loading spring thrust washers to take up shaft end-play, or had them on the wrong end. In the case of bulkhead-mounted equipment which had to slant forward, if the spring washer was at the rear, and in the case of vertically-mounted synchros in computors or range-keepers, if it was at the bottom end, the weight of the rotor compressed the spring thrust washer and the synchro rotor would hum and vibrate. This was a distracting nuisance in the case of vital ship stations under combat area conditions. Such synchros should have the thrust washer at the front end, or at both ends of the shaft.

A good deal of test and maintenance troubles with synchro and synchro-controlled servos was caused by a lack of explicit and unambiguous zeroing information procedures in the operating and instruction manuals supplied with the military gear. Four distinct zeroing operations usually had to be carried out in the case of most systems—

1)—Both the remote-control and follow-up equipment had to be set exactly in their mechanical zero positions.

2)—Both units' geared indicating dials had to be set to read zero.

3)—Both the units' control and indication synchros had to be set in their electrical zero positions, with their dials indicating zero.

4)—Both units' synchros had to be checked to make certain that they put out an increasing-reading signal when either unit was actuated in the increasing-reading direction.

Failure to observe such accurate and complete zeroing routines, also resulted in confusing rounds of dial shifting, synchro resetting and wire reversing, especially in the case of servo drives. This in turn, sometimes resulted in further chains of reversals and resettings when other systems were cross-tied in.

The necessary use of some differential synchros in military equipment has complicated system design somewhat. An interesting thought here, might be a 6-phase-type star-wound stator synchro which might be able to combine two synchro signals in a differential fashion electrically.

SYNCHRO TRANSMISSION LINE AND POWER SUPPLY CIRCUITS

Most synchro systems use a simple blown-fuse indicator for the rotor supply fuses. This usually consists of a neon lamp across each fuse, the idea being that when a fuse blows, the line voltage appears across the open fuse and lights up the 115-volt neon bulb. However, in the case of synchro systems where each synchro has its own rotor supply fuses and blown-fuse indicators, the warning circuits failed to operate. Since the open-circuit rotor synchro was being fed by the transmitter's Y-circuit signal which induced 115 volts back into its "dead" rotor, both sides of the open fuse carried an in-phase 115-volt potential and the neon lamp across the blown fuse could not light up. A similar condition would exist in the case of a paralleled bank of transformers. This was a fooler which caused some 180-reversed-indication and burned-out synchro troubles.

Where different sections of cross-tied synchro systems had to be fed by supply voltages of somewhat different sources, some peculiar operational troubles developed when the supply voltages were not exactly of the same voltage, phase and frequency. This condition sometimes cropped up in the case of radar and thyratron-servo F.C. gunnery equipment where 2- or 3-phase transformers

or motor-generator sets were involved in the power supply system. If the rotor supply voltage fed to a synchro generator was not the same as that fed to one of its indicators, the two sets of Y signal voltages would not be the same at static synchronism, although their phasing and ratios would. Hence a steady Y-winding circulating current would flow at static synchronism and the two units would gradually overheat. Again, this was a condition similar to that of paralleled banks of transformers whose primary or secondary voltages were not exactly the same. If the system had trouble-light circuits, they would light up even though there actually was no trouble in the synchro system itself, since Y circuit currents were flowing at synchronism. If the phases of the two rotor supplies were not the same, the indicator synchro rotor would indicate an angular error equal to the phase difference between the two rotor supplies. (If the rotor leads are reversed at the indicator synchro, for instance, the power supply is in effect reversed 180 degrees in phase and the indicator snaps into a position 180 out with respect to the transmitter.) If the two power supplies differ slightly in frequency, the indicator motor will go into a sharp oscillation whose frequency will equal the difference frequency between the two supplies. Hence great care must be exercised in designing a synchro system to make certain that all synchros in a given system are fed by the same power source. These peculiar operational troubles might suggest some possible special commercial applications for synchros.

Where excessively long cable runs existed, as in the case of 880′ battleships, for instance, other synchro and servo operational troubles sometimes cropped up. Long cables possess some inductive reactance, especially when running along steel bulkheads. Their reactive surge currents tended to make synchro indicator motors unstable and liable to hunt and motorizing tendencies. This indicated the need for p.f.-correcting delta banks of condensers across the long Y-circuit lines, and possibly, heavy condensers across the rotor-supply lines. In the case of long control-synchro rotor lines, stray A.C. signals were sometimes picked up by induction and fed as false signals into servo amplifiers to cause some jitter and creep effects at synchronism. This indicated the need for twisted-pair and shielded conductors and cables, or voltage-bucking circuits in long control-synchro rotor lines.

Where flexible-cable loops were used to feed a remote-control head or follow-up unit with a limited-rotation sweep, not only some attention had to be paid to the cable sheath thermoplastic characteristic when the equipment had to operate under very high or very low ambient temperature conditions so as to guard against cold-splitting and heat-flow effects, but also to conductor wax and oil impregnant migration-flow in hot-climate areas. Care also had to be taken to prevent the cables from rubbing or striking against any obstructions or bulkheads, even slightly, otherwise the cable sheaths wore through rapidly and shorted, grounded out or opened up the cable conductors. The cable conductors also had to be sufficiently stranded and flexible enough to withstand the constant twisting and flexing actions without developing premature fatigue-crystallization failure. This type of trouble was apparently hastened by temperature cycling and vibration conditions common to most military equipment. The heavy insulation spacer blocks used to space the flexible cables, also contributed to this trouble by introducing localized flex points on both sides of the blocks. Spacer blocks made of soft synthetic rubber or plastics, or sliding laminated blocks with deeply chamfered holes, would also help out here.

Where limitless rotation of the remote-control and servo-driven units was necessary (as in the case of A.A. gear) and slip-ring assemblies were required, some sort of provisions sometimes had to be made to insure that the slip-rings and their brushes would not be contaminated by the servo drive's lubricant and hydraulic servo oils, or by sea-water, otherwise slip-ring dead-spots and erratic servo operation would obviously result. Where moisture, explosive fumes or chemical vapors were present in excessive amounts, these assemblies had to be sealed off in some makeshift fashion to prevent chemical and electrolytic corrosion effects at the slip-ring components.

THE SYNCHRONIZING RELAY CIRCUITS

Some precision servo drives employ dual or triple-speed synchros where the geared hi-speed or "fine" units take over control at close to synchronism. The vernier control-signal reduces synchronism errors caused by synchro and wiring unbalances and mechanical backlash in inverse proportion to the gear ratio. Some servo systems employ synchronizing relays to switch control from the low-speed or "coarse" synchro over to the hi-speed vernier. Where the usually trouble-free vacuum relays were used, trouble sometimes developed from fatigue-crystallization cracking of the armature pivot diaphragm or bellows, air leakage, and rapid failure of the contact points. Another problem sometimes cropped up with a peculiar resistance "coating" of the vacuum-protected contacts after long stand-by periods of idleness. This could sometimes be burned clear with heavy current pulses. While these troubles were mostly noted in the case of older servo designs, it may be that better diaphragm alloys and the cause for vacuum-protected contact-point coating might be worth investigating.

Regardless of the type of relay used, it should have some provision for contact-point gap adjustment and contact-point armature tension control. In this way, the contact-point gap adjustment can be maintained in spite of arc erosion, and occasional file dressing in the case of sealed units. Hence the armature pull-in and drop-out points can be maintained for proper servo operation with relay amplifier tube changes and aging and wearing of the relay mechanism. Where a relay amplifier tube circuit is used, a cathode or plate rheostat or potentiometer control is useful for adjusting relay sensitivity to correct for various mechanical and electrical variables. This eliminates the need for constant relay-amplifier tube checks and replacements.

An auxiliary chassis-mounted test switch cuts out the synchronizing relay and switches servo control into only either coarse of fine control at will, is a useful addition for speeding up test and maintenance routines.

THE VOLTAGE AMPLIFIER CIRCUITS

Regardless of whether the voltage amplifier be a D.C., phase-sensitive A.C., or a conventional amplifier, chassis-mounted screw-driver-slot rheostats or pots should be provided for balancing up both halves of p.p. or full-wave amplifiers in each stage, otherwise various tube and circuit unbalance variables will result in some residual amplifier output even though a "zero" static-synchronism control-signal is present. This may result in some sync error and jitter and

creep. Such balancing pots save much fuss and bother with constant checking and matching and pairing up of tubes in each stage at regular intervals.

The servo amplifier gain requirements are largely determined by servo response speed, accuracy and drive-power requirements, hence variations in amplifier gain will have an effect upon servo response. If the power supply voltages come up, or tube plate currents and amplification factors are above normal, the servo response may be too "hard," with too much over-shoot, chatter and weave. If below normal, the reverse will occur and servo response will be too "soft" and too sluggish since amplifier gain will be below normal. Similarly, servo response will be affected by ambient temperature extremes. Under very cold operational conditions, the servo drive's mechanical linkages and bearings may develop considerable shrink-fit and lubricant-thickening viscosity friction effects. This will obviously dampen down servo response. The reverse will occur in very hot weather. For these reasons, some sort of a voltage-amplifier gain adjustment is advisable for military servos. This may be obtained by means of chassis-mounted screwdriver-slot pots which control the input signal or voltage-amplifier stage gain, or by a tapped-primary input transformer. To some extent, suitable inverse-feedback networks may correct this type of trouble.

In checking the relative phasing and polarity of synchro control signal, servo amplifier, servo motor, saturable reactor, amplidyne, rate generator and power supply voltages, field engineers and test technicians sometimes had to resort to some acrobatics in order to take the necessary readings. This was particularly troublesome on the case of superstructure gear such as searchlights, gun directors, radar antenna arrays, A.A. machine-gun cannon, etc., particularly during inclement weather conditions when precision equipment was exposed to rain, snow or dusty winds. If the equipment terminal boards were so laid out that all these vital voltages could be readily accessible (especially the control synchro rotor signals), this time-consuming troublesome maintenance problem would be considerably simplified. Such time-saving test-points (T.P.s) are standard practice in many types of electronic equipment designs and ordnance design men might take a page out of the electronic engineer's book here with some benefit.

In laying out the amplifier equipment, some thought might be given to the fact that most military gear is subsequently modified or added to in some way because of field operational or combat experiences. The costs of such modifications would be greatly reduced if the lay-outs were so made as to provide for spare terminals, extra cable holes, spare wires, spare stuffing tube holes plugged up with pipe plugs, etc. While this increases the cost of equipment some initially, it is far cheaper in the long run since such modifications on existing designs are difficult and laborious.

Where dual or triple sets of servo drives are required (bearing-elevation-range combinations, for instance), it is of some advantage to work out over-all system design so that identical amplifiers can be employed. This can usually be done by reducing the control-signal amplitude, cutting down the anti-hunt rate signals, and reducing servo motor size for the lighter-load servos. This simplifies test, maintenance and rear-echelon supply work since only one type of servo amplifier need be stocked and serviced, and can be readily swapped around in the field to quickly determine whether servo system trouble is located in the amplifier itself, or elsewhere.

THE POWER AMPLIFIER CIRCUITS

Where more massive servo drives involving heavy motors and large thyratron tubes were concerned, peculiar troubles were sometimes encountered with erratic behavior of these big tubes, in spite of shock mounts and thermostatically-controlled ovens. It appears that the effects of strong radiation, magnetic and electric fields; actinic light; and barometric pressure variations upon large thyratrons' firing points might be worthwhile investigating in some detail. This may be of some significance in the case of hi-velocity hi-altitude rocket telemetering applications, even though small thyratrons are used here for the PM radio links.

As in the case of voltage-amplifier stages, a balancing pot of the chassis-mounted screwdriver-slot type should be provided to balance up the power amplifier tube plate currents to eliminate the need for checking and matching of these costly tubes.

If magnetic power amplifiers are used (saturable-core reactors or transformers), field work would be simplified if all the windings are carefully wiring-coded so that the relative phasing and polarity of both the A.C. and D.C. windings are established with respect to the control signals and power-supply voltages. This simplifies test work and prevents the familiar "two-wrongs-canceling" condition which makes for continual maintenance troubles. The nominal D.C. ohmic resistance of all reactor windings should be stamped upon the unit's nameplate and indicated on the wiring schematic to speed up cold-circuit ohmmeter checking. This also applies to all synchros, transformers, reactors, rate generators, relays, amplidynes, servo motors, motor-generator sets, etc., in general.

Where a hydraulic or pneumatic power-amplifying stage is present, some provision for tightening up on gearing and linkage wear backlash at the valve actuating mechanisms should be provided to prevent servo sync error and jitter and creep.

If heavy D.C. servo motors are being directly driven by amplifier power tubes, full-wave circuits may be worthwhile in the case of more massive servos since the higher ripple frequency and smoother D.C. allow the use of smaller filter chokes and more conventional D.C. servo motor designs.

THE SERVO MOTOR CIRCUITS

Servo motors of the commutator types should have readily removable bell-housing cover plates for rapid visual inspection of the brushes and commutator. Greasy or pitted commutator bars, or gummy and worn brushes may result in weak and erratic servo response, or in servo motor "dead spots."

A chassis-mounted test switch and a heavy "dummy load" resistor circuit would be useful for shunting out the A.C. windings of saturable-core reactors to trigger off the servo motors. This would simplify servo motor testing and help isolate servo operational troubles. A gear-shift or an "in-out" clutch mechanism for disengaging the servo motors from the follow-up units for free-running tests, would also help in quickly isolating operational troubles.

Where servo motors employ alnico permanent-magnet fields, both magnets and mounting frame should be plainly stamped with N and S polarity designations so that magnets will not be reversed during routine field mainte-

nance checks. Such simple mix-ups have resulted in whole servo systems being taken apart to discover why they were running away, or running in the wrong direction. Off-set bolt-holes or locating pins would serve much the same purpose. Much the same holds good for rate generators and amplidynes.

In specifying the directions of rotation for rotating equipment, the terms "right-hand" and "left-hand" should be avoided since they are ambigious. If the terms "c.w." or "c.c.w." are used, the direction of sight should be stated plainly to avoid confusion. For instance—"The amplidyne should rotate in a c.w. direction, looking at the motor end." In general, the terms "increasing-angle direction" or "decreasing-angle direction" are more reliable since two different servo drives on the same equipment may rotate in opposite directions for the increasing-angle directions. Where directions of rotation are indicated on a motor or generator, it is usually better to simply show an arrow marker than to rely on confusing words.

THE ANTI-HUNT RATE-COMPONENT CIRCUITS

Chassis-mounted screwdriver-slot pots should be provided to allow for some adjustment of the amount of anti-hunt rate signals fed back into the servo amplifier (velocity and acceleration rate components), with changes in power supply voltages, tube characteristics, ambient temperature shifts and servo load conditions for optimum servo performance under all operational conditions. Factors causing an increase in amplifier gain and a decrease in servo loading and damping, also usually make a stronger set of anti-hunt voltages necessary in order to prevent hard servo response and excessive over-shoot, weave and chatter. Rate generator windings should be wiring circuit-coded carefully so as to indicate relative phasing and polarity against the rest of the servo system. Reversed rate circuits will cause a servo to run away hog-wild in the case of the first-derivative velocity-rate signal, and to oscillate or chatter violently in the case of the second and higher-derivative acceleration-rate signals. Here again, servo drives have been dissected for sync troubles caused by rate-circuit reversals in the external rate and amplidyne-generator field circuits.

THE POWER SUPPLY CIRCUITS

If 3-phase or split-phase A.C. power supply circuits are involved in a servo system, a simple phase-rotation indicator of some kind should be provided at the power-supply control panels and servo amplifier chassis. This could consist of a simple unbalanced-Y light circuit, or an induction-motor gimmick of the hysteresis type. Mixed-up A.C. power leads' phasing caused considerable confusion and operational troubles in the case of phase-shift grid-control thyratron servos. Such mix-ups completely deranged the thyratron servo circuits which no amount of reversing and swapping of wiring elsewhere in the system would correct for in a "four-wrongs-make-a-right" fashion. Power thyratrons function as grid-controlled rectifiers, and the firing and extinction points of these tubes are fairly critical as to control-signal phasing and voltage.

All D.C. and A.C. power supply leads in a servo system should be circuit coded in such fashion as to indicate relative phasing and polarity against the rest of the servo system's circuits, for reasons already discussed.

THE MILITARY DESIGN-MAINTENANCE COORDINATION NEED

In conclusion, it might be pointed out that during World War II, experienced military personnel were often aware of certain field operational shortcomings of some of their equipment. Others had devised ingenious makeshift modifications which increased their effectiveness considerably. Yet, paradoxically enough, little or nothing could be done about either one or the other, outside of going in a long and roundabout fashion through the National Inventors' Council in the case of a major item. It would seem that some sort of a "beneficial suggestion" system for military personnel could be worked out which could be similar to the one the Navy adopted for Navy civilian personnel during the war with such great success. Ideas investigated and accepted by appropriate military agencies, could be rewarded by extra points on rate promotion exams, and reduction in the service time requirements for routine promotions. Similarly, field service reports similar to the ones used by the large automotive concerns, might also be used effectively to complete the research-design-production-maintenance chain to ultimately result in better designed military equipment for our armed services.

(Note:—These comments have been abstracted from a series of test instruction bulletins and field test reports prepared by the writer while with the Electric Shop Ship Repair Force Test Section at the N. Y. Naval Shipyard, Brooklyn, N. Y., and from an unpublished work on selsyn systems.)

STUDY YOUR CENTRIFUGAL PUMPS BEFORE PARALLELING

ACKNOWLEDGMENT

This article discusses the use of pump characteristic curves for safe paralleling of centrifugal pumps. Written by ROY CARTER and IGOR J. KARASSIK of Worthington Pump and Machinery Corp., it was published in the April 1952 issue of "Power."

Pumping systems fall into (1) throttled and (2) unthrottled. In (1), flow is controlled by throttling with one or more valves the excess head developed by pump or pumps. On some systems, like boiler-feed, a throttle valve directly in the discharge line controls flow. In others, such as city-water supply systems without a standpipe or reservoir floating on the distribution mains, the water users control pump discharge as they open or close valves to suit their needs. In (2), unthrottled systems, where pumps discharge into a standpipe or reservoir, flow depends on head they develop and normal characteristics of system. Here, we will consider throttled systems only.

Boiler-Feed Pumps. Fig. 3 is a simple boiler-feedwater hookup whose flow is controlled by throttle valve *A*. This valve can be positioned by hand or automatically by a feedwater regulator. Fig. 1 shows the feedwater-system head curves superimposed on the pump's head-capacity curve. On these curves line *CB* represents boiler pressure plus the static pressure in the pumping head.

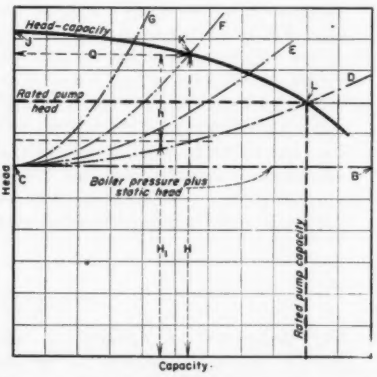


FIG. 1 — Boiler-Feedwater system head curves plotted on feed pump head-capacity curves show capacities pump runs at.

This pressure is practically constant varying only with slight changes in boiler pressure.

When water is supplied the boiler by the pump, pressure against which it operates increases because of friction-head losses in piping, fittings, valves and other devices in the line. With valve

A, Fig. 3, open, we can assume that pressure due to friction losses increases along curve *CD*. This curve crosses the pump head-capacity curve at point *L*, which is the rated head and capacity of pump.

Partly closing valve A increases friction head and system's head curve might rise to position *CE*. Further closing of valve *A* would produce other head curves like *CF* and *CG*. If valve is closed completely, pump pressure goes to shutoff at *J*. Then, by opening valve *A*, head curve can be varied so it appears as a family of curves, representing a fully closed valve at *J* and an open valve at *D*.

With head-capacity curves for a boiler-feed pump as in Fig. 1, to supply the boiler with a quantity of water, *Q*, throttle valve is adjusted until system's head curve is *CF*. This curve crosses the head-capacity curve at *K* and head against which pump operates is represented by vertical distance *H*. Actual head required to deliver quantity Q to boiler on normal head curve *CD* is represented by H_1. As pump develops head *H* at capacity *Q*, valve *A* will have to throttle away a head equal to H-H_1 or *h*.

Rising Head Curves. When two like pumps with steadily rising head-capacity curves operate in parallel, system load should divide equally between them. Fig. 4 shows the head-capacity curve of either of two like pumps. To draw this curve for the two pumps operating in parallel we need only project capacity points at various heads on the curve to the right to double their value. For example, point *A*, 1000 gpm at 244-ft head, is extended to the right to 2000 gpm and 244-ft head, point *B*. The 2000-gpm intersection *C* is projected to 4000 gpm, point *D*, and in like manner 3000 gpm is projected to 6000, and 4000 to 8000 gpm, as horizontal dotted lines indicate. Curve drawn through these points is head-capacity characteristic for the two pumps. Each pump is rated 3500 gpm at 197-ft head, point *E*, so two pumps are good for 7000 gpm and the same head, point *F*, on the 2-pump curve shown in Fig. 4.

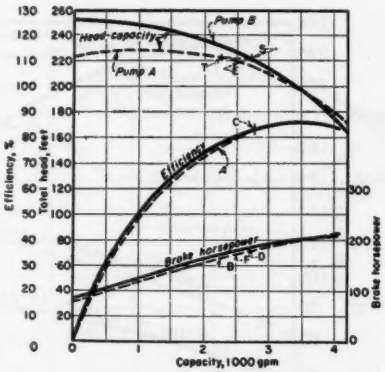


FIG. 2—Two pumps with unlike head-capacity curves may operate in parallel but problem is different from one for like units.

Power input to the two pumps is about 400 hp, point *G* on the horsepower curve for two pumps. This curve

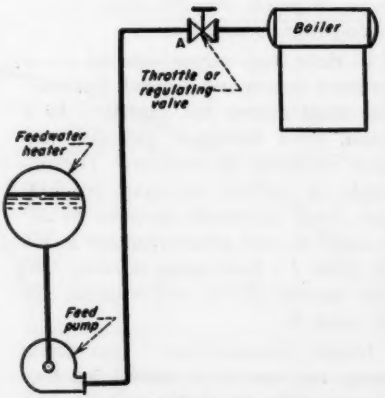


FIG. 3—Simplified diagram of a boiler feedwater system hookup with a manually or automatically controlled valve at A.

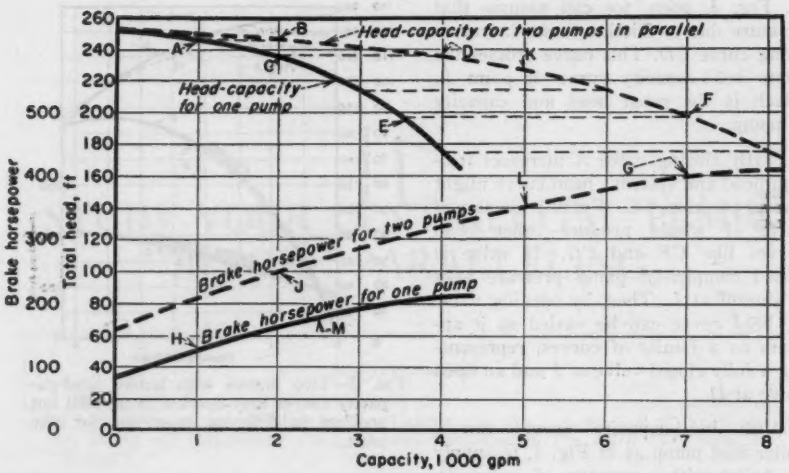


FIG. 4—Head-capacity and brake-horsepower curves for two like pumps and combined curves for the two pumps in parallel.

was plotted by doubling the horsepower for one pump and doubling the capacity. For example, at 1000 gpm, the power is 125 hp for one pump, point *H*. So for two pumps it is 250 hp at 2000 gpm, point *J*. Repeating this procedure at different capacity values we get a series of points through which the 2-pump power curve is drawn.

If these two pumps operate on a throttled system and demand decreases, they must deliver less capacity. As a result, head developed increases and more throttling is necessary. For example, if demand decreases to 5000 gpm, head developed increases to 227 ft, point *K*, and power required is 356 hp, point *L*. Each pump delivers 2500 gpm against 227 ft and requires 178 hp, point *M*.

Unlike Characteristics. Two unlike pumps can operate in parallel, but they present different problems from two that are alike. Let's study the two units whose characteristics are shown by full-line and dotted curves, Fig. 2. Either can be used alone on a throttled system. If power cost is high, pump with dotted curve would be preferred because its power at part capacities is slightly lower.

Don't infer that of two pumps the one with flatter head-capacity curve takes less power at part capacities, because this is not always true. Input depends on impeller and casing design, relation of design point to maximum efficiency, and other factors. So a pump with a steep head-capacity curve may take less power at part capacities than one with a flatter curve. But a unit with a steep head-capacity curve, full line Fig. 2, has the advantage in a single-pump throttled system requiring a greater travel of the throttling valve, because more head has to be lost across valve. Operation is more stable.

Drooping-Head Curve. In spite of having a drooping head-capacity curve, dotted line Fig. 2, this pump can usually

be used on a single-pump throttled system as operating point is determined by throttle-system head. On some throttled systems with a single pump having a drooping head-capacity curve, flow surging may occur. This has also developed to a less degree with pumps having a stable, constantly rising head-capacity curve, full line Fig. 2, but such cases are rare.

When two or more pumps operate in parallel, Fig. 5, each has its own piping and fitting losses as well as losses common to both. Instead of using the true head-capacity curve, use one measured from *A* to *B* in system study. This is the true characteristic of the pumps minus piping and fitting head losses from *A* to *B*. Usually in throttled systems, particularly in high-head applications like boiler-feed, losses in each pump's piping are such a small part of total head, they may be neglected.

Combined Head Curves. Fig. 6 shows characteristic curves of two pumps,

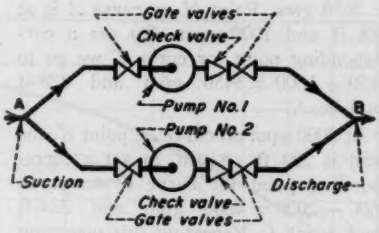


FIG. 5—Simplified piping hookup for two centrifugal pumps run in parallel.

each designed for 3500 gpm, 197-ft total head, point *N*, but with different-shaped head-capacity curves. *A* and *B* are head-capacity curves of the two pumps; *C* is the combined head-capacity curve. This curve includes values from *D* to *E* on curve *B* to which is added curve *A*, and extends from *E* to *F*. Point *G* on curve *A* is at 0 gpm, while *E* on *B* is at 2650 gpm, with 223-ft head on both curves. Combined capacity of the two pumps at 223-ft head is 0 + 2650

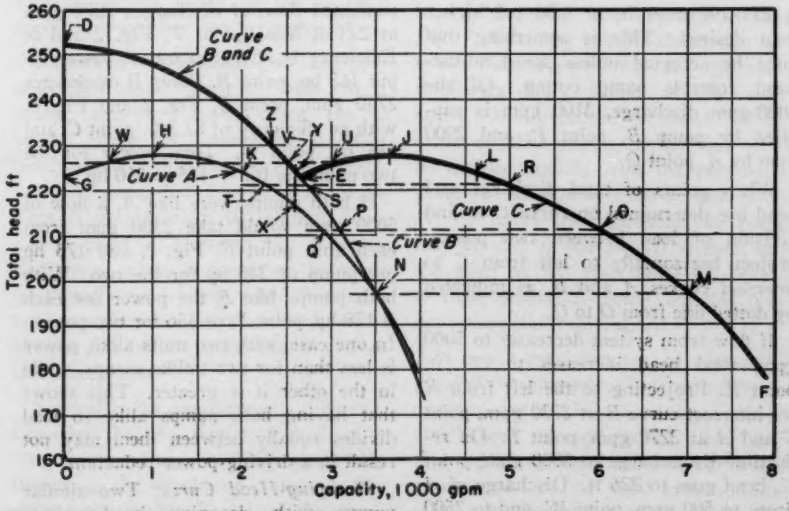


FIG. 6—Characteristic curves, individual and combined, for two pumps, each designed for 3500 gpm at 197-ft total head, but having different head-capacity.

= 2650 gpm. Point *H* on curve *A* is at 228 ft and 1000 gpm. To get a corresponding point for curve *C* we go to 2450 + 1000 = 3450 gpm and 228-ft total head.

At 2000 gpm on curve *A*, point *K*, the head is 224 ft. Again, to get a corresponding point for curve *C* we go to 2600 + 2000 = 4600 gpm and 224-ft head, point *L*. Repeating this operation we get a series of points through which curve *C* can be extended to represent the combined characteristics of the two pumps. Construction of this curve is the first step in finding how two pumps with unlike head-capacity curves perform on the same system.

Waterworks Plant. Assume these pumps are in a waterworks plant discharging directly into a distribution system at a normal total head of 197 ft. At this head the two pumps discharge 7000 gpm, point *M*, where both have the same capacity, point *N*. If flow in system falls to 6000 gpm, point *O*, the two pumps develop 211-ft total head. Under this condition, pressure in main is 211–197 = 14 ft, or 6.06 psi higher than desired. This is something that must be accepted unless speed adjustment controls pump output. Of the 6000-gpm discharge, 3100 gpm is supplied by pump *B*, point *P*, and 2900 gpm by *A*, point *Q*.

When points of total discharge and head are determined on curve *C* to find division of load between two pumps, project horizontally to left from *C* to intersect curves *A* and *B*, as indicated by dotted line from *O* to *Q*.

If flow from system decreases to 5000 gpm, total head increases to 221 ft, point *R*. Projecting to the left from *R* we intersect curve *B* at 2730 gpm, point *S* and *A* at 2270 gpm, point *T*. On reduction of discharge to 3000 gpm, point *U*, head goes to 226 ft. Discharge of *A* drops to 500 gpm, point *W*, and to 2500 gpm on *B*, point *V*.

If system head is throttled to 230 ft, discharge drops to 2350 gpm for pump *B*, point *Z*. *A* won't discharge because system head has increased above the maximum 227 ft, point *H* for this unit. Pump *A* is therefore "backed off the line" by *B*, and operates at shutoff, a dangerous condition even for a short time. The two pumps should not operate in parallel at a capacity below a safe minimum.

Load Division. Assume *A* operating alone and discharging 2600 gpm, at 216-ft head, point *X*, when *B* starts. The latter has a head-capacity curve higher than *A* and picks up the load to back *A* off the line. If flow in system stays constant at 2600 gpm, the head rises to 224 ft, point *Y*. On the other hand, if *B* had been discharging 2600 gpm alone and *A* had started, *A* could not have supplied any water to the system.

Suppose these pumps operate on a system where the demand changes slowly and one unit is taken out of service when the load falls below 3500 gpm. Under careful supervision the two could operate safely in parallel. At 5000 gpm combined flow, *A* discharges 2270 gpm at 221 ft head, point *T*, Fig. 2 and 6. Efficiency is 76.8%, point *A*, power input 165 hp, point *B*. Pump *B* discharges 2730 gpm, point *S*, Fig. 2 and Fig. 6, with an efficiency of 82.3%, point *C*, and 185 hp, point *D*. Total power for the two pumps is 165 + 185 = 350 hp.

If both pumps were like *A*, a flow of 5000 gpm would take 2500 gpm from each unit, point *E*, Fig. 2, and 173 hp per pump or 346 hp for the two. With both pumps like *B* the power for each is 178 hp, point *F*, or 356 for two pumps. In one case, with two units alike, power is less than for two unlike pumps, while in the other it is greater. This shows that having both pumps alike so load divides equally between them may not result in a driving-power reduction.

Drooping-Head Curve. Two similar pumps with drooping head-capacity characteristics, dotted curve, Fig. 2, can

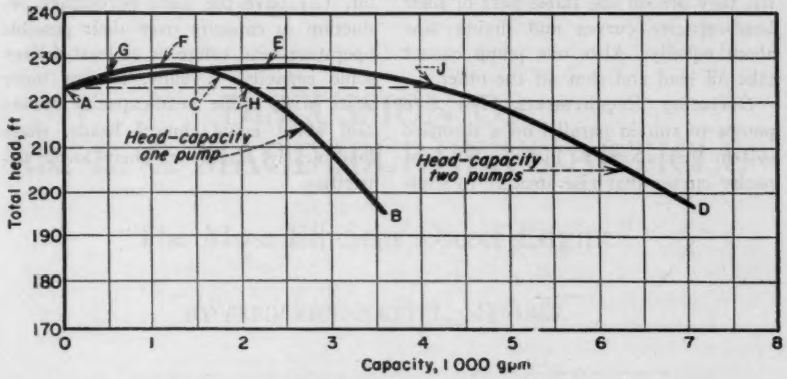


FIG. 7—Two similar centrifugal pumps having drooping head-capacity characteristics can give operating trouble when connected in parallel on a throttled system.

give trouble when operated on a throttled system. Curve *AB*, Fig. 7, shows head-capacity characteristic of either of two like pumps operated in parallel. Curve *AD* is theoretical combined head-capacity relation of the two. This curve is developed as explained in Fig. 4, assuming friction losses so low in each pump's piping they can be neglected.

Assume a water demand of 1750 gpm and 226-ft head, point *C*, with No. 1 pump operating alone, when No. 2 starts. Pump No. 1 operating at 226-ft head applies this pressure to check valve of pump No. 2 to hold it closed. This is greater than the shutoff head, 223 ft, point *A*, of pump No. 2. Thus, this pump when it comes up to speed on shutoff, can develop only 223-ft head, not enough to open the check valve against 226 ft, and it cannot discharge into the system.

Several tricks overcome this difficulty. Consider the two pumps, Fig. 5, with No. 1 operating at 1750 gpm. By throttling the discharge valve of this pump, pressure at *B* and on the check valve of No. 2 can be reduced below the shutoff head of this pump. Then, when it is started it can open check valve and take part of the load. This and other tricks to get second pump on system generally require careful handling of valves and timing.

Division of Flow. With two pumps having drooping head-capacity curves, Fig. 7, flow may not divide equally between them even if they are hydraulic duplicates and operate at the same speed. For example, to supply a demand of 2250 gpm, point *E*, load could be divided equally with the two operating at 1125 gpm and 228-ft head, point *F*. Or under this condition one pump may operate at point *C*, 1750 gpm, and the other at 500 gpm, point *G*. Heads at these points are equal, but this does not insure stable operation.

It is not good practice to operate in parallel two pumps with drooping head-capacity curves, Fig. 7, at capacities at which head exceeds shutoff. In Fig. 7 this is 2100 gpm and 223-ft head for one pump, point *H*, or 4200 gpm and the same head, point *J*, for two. When the two operate at this or a greater capac-

ity, they are on the stable part of their head-capacity curves and divide flow about equally. Also, one pump cannot take all load and shut off the other.

Operating Requirements. For two pumps to run in parallel on a throttled system they should: (1) have head-capacity curves that rise steadily to shutoff, (2) have the same percentage reduction in capacity over their possible operating-head range or at least deliver some capacity. Usually, two or more units with stable head-capacity curves and about equal shutoff heads, share load nearly equally at their lower capacities.

DISCUSSION OF MR. E. C. MAGDEBURGER'S ARTICLE

"The Most Efficient Diesel Engine"

BY FREDERICK NETTEL, MEMBER

DISCUSSION

In the February 1952 issue of the JOURNAL MR. E. C. MAGDEBURGER presented "The Most Efficient Diesel Engine." Printed below are a discussion by Mr. Frederick Nettel and Mr. Magdeburger's reply. Following that exchange of views we publish the engineering and test data of the engine under discussion as presented by Drs. Pflaum and Eichelberg.

It will be very interesting indeed to have more information on the MAN tests.

It seems, however, desirable even at this moment, to comment on Mr. Magdeburger's remarks in order to reestablish a proper perspective:

Precooling of the air before it enters the compressor (see page 170 near bottom), to have an appreciable effect, is obviously predicated on the availability of a cooling medium very substantially lower in temperature than the ambient air. It will be interesting to learn whether such a cooling medium is available or can be made available on board ship, or for that matter in most other applications.

Thus most of the cooling will have to be done during and/or after compression. The effect of both were indeed not only thermodynamically predictable but well known for decades from practical engine application. Such cooling is practiced so far mostly on larger engines.

Also the shortcomings of the Buchi system have been recognized for many years, but it remained unchallenged until turbocharged sets of much higher product efficiency became available.

MAN accepts what are considered today very high peak pressures for no other reason than "that the reliability of ignition by compression must not be jeopardized, especially during starting."

These high pressures admittedly cause the adoption of a crosshead design which materially increases engine height, and larger bearings. Both must increase the specific engine weight substantially.

The question arises: are these high pressures with the mentioned consequences unavoidable?

The answer, on the basis of the presently known state of art is NO.

It has been shown both to the industry and to the Navy that it is possible to design an engine with compression ratios much lower than that chosen by the MAN (even down to 9:1 or less)

and still start it reliably. It can be reasonably predicted that starting of such an engine will be possible even at abnormally low ambient temperatures.

This is done by providing in an engine designed with a compression ratio too low for compression-ignition, a simple auxiliary combustion chamber in a by-pass pipe between the compressor discharge pipe and the exhaust gas turbine inlet, which permits the turbocharger set to be operated prior to starting the engine as an independent gas turbine set supplying precompressed and therefore suitably preheated air to the cylinders. This combustion chamber need be operated only for short periods so that the economy of the engine is not harmed.

Such a turbocharger set can even be used for cranking the engine.

The MAN—or perhaps Mr. Magdeburger only—seem to make a virtue of the high compression ratios and ascribe the superior performance of their engine to them. Prof. Pflaum himself has, however, shown that the theoretical cycle efficiency for the mixed combustion cycle chosen by the MAN rises very little indeed for compression ratios in excess of 20:1 and falls by about as much with falling air intake temperature. (See Pflaum "IS-Diagramme fuer Verbrennungsgase" pp. 24 and 29.) Any increase that may be left is likely to be eaten up by the deteriorating mechanical efficiency.

At best only a portion of the improvement in fuel consumption can thus be accounted for by the increase in compression ratio; the rest must be due to better combustion. MAN also utilizes to good advantage the recent great advances in compressor and turbine design (sparked by gas turbine developments) which result in a turbocharger set capable of furnishing the high density charging air with comparatively low turbine inlet temperatures.

On the basis of experimental work on much smaller engines it can be stated, however, that similar results are obtainable without the penalties in pressures, and consequently weight and space, involved in the MAN design, by adopting the system proposed by the writer. This system gives complete freedom in the choice of top pressures while ensuring not only equal but superior starting performance.

Mr. Magdeburger's criticism of the U.S. Diesel builders seems justified even though he cites engines of rather old vintage. It is not quite clear what the Nordberg gas engine has to do with the problem under discussion.

MAN's clear thinking and courage in developing this engine deserves credit, especially if we remember that there is no government which can be asked to foot the bill. MAN's forthcoming report will be read with great interest by all concerned and it must be hoped that it will reawaken interest in similar developments going on in this country.

In his concluding remark Mr. Magdeburger displays a redeeming sense of humor by proposing a marching formation which will give food for thought to the best military minds: He suggests to stay "ahead of the rest of the world" —while remaining all the time carefully behind Germany!

In reply to Mr. Nettel:

Mr. Nettel's comments upon my attempt at evaluating the new MAN high-pressure turbocharged four-cycle engine present a welcome opportunity for further discussion of results obtained in testing this milestone of progress.

First it must be noted that discussion of exhaustive tests of a 1400-horsepower engine is different from Mr. Nettel's presentation of claims made on the basis of experimental work on much smaller engines. The first has reached the "proof of the pudding" stage while the latter

merely reflect the enthusiasm of the inventor. Actual test data of the engine have been published in the German VDI ZEITSCHRIFT of 21 Dec. 1951, p. 1113 and an extract in the "British Motor Ship" of Apr. 1952, p. 18 and will be published in full in this JOURNAL.

The principal thing to remember is that a given cylinder volume has been made to yield three times the power of conventional atmospheric induction engine and in addition proved to be "the most efficient." The price paid for it by MAN is higher peak pressures, that is quite true, but Mr. Nettel questions even the availability of the cooling medium—actually the Augsburg water was quite adequate.

Mr. Nettel's assurance that high peak pressures are avoidable and his prediction that engines with compression ratio of $9 \div 1$ or less can be started "by adopting the system proposed by Nettel" need not be doubted. Eventually means will be developed to accomplish both, but at the moment MAN chose the simplicity of adequately high compression ratio to do it.

High compression ratios have not been discussed in my article and therefore no undue virtues have been claimed for them by me. If anything the compression ratio of the new engine is as low as would provide starting ignition reliably and simply and it actually is below the conventional figures now used.

I rejoice in the low fuel consumption demonstrated by the engine and believe it is the result of many causes named in the Eichelberg-Pflaum article. But, as stated above, the main thing is—it is the lowest ever—practical engineers are not likely to quarrel about the reasons why. Furthermore recent proposals to the Navy have shown that the results obtained by MAN have already had their beneficial effect on the engine industry in this country.

The new Alcoa plant at Lavacos, Texas, one of the largest in the world, supplies power for production of aluminum at lowest possible cost, yet its BMEP rating is only about one-fourth of that demonstrated by the new MAN engine. Is that not sufficient reason for including it in the comparison table?

Finally may I ask Mr. Nettel what is wrong with the invitation for "the Navy making use of the new engine ahead of the rest of the world"? Does it mean "remaining behind Germany"? If so, let us make use of the new engine anyway.

E. C. MAGDEBURGER.

HIGH PRESSURE CHARGING OF DIESEL ENGINES TO INCREASE OUTPUT AND REDUCE FUEL CONSUMPTION

ANALYSIS OF TEST RESULTS OF A NEW MAN FOUR-CYCLE ENGINE

ACKNOWLEDGMENT

A paper was initially prepared by Professor Dr.-Ing. G. Eichelberg of the Federal Technical University of Zurich in Switzerland and Professor Dr.-Ing. W. Pflaum of the Technical University Berlin-Charlottenburg in Germany and presented by them before a German Technical Society meeting in December 1951. A somewhat abbreviated article based on the same material was published by the authors in the VDI Zeitschrift of 21 December 1951 and by W. Pflaum in MTZ-Motortechnische Zeitschrift Number 2 (1952). The present draft was especially prepared by the authors for the JOURNAL. It was translated from the original German by Mr. E. C. Magdeburger, a frequent contributor to the JOURNAL and an outstanding Diesel engineer in the Bureau of Ships, U. S. Navy Department.

BASIC CONSIDERATIONS FOR INCREASE OF OUTPUT

The concept of output increase, generally speaking, refers to an engine-unit and to begin with, it is immaterial how it is done. The various possibilities are clear from the study of the basic equation of brakehorsepower of an internal combustion engine

$$N_e = C \cdot V_h \cdot z.n.p_e$$

(European equivalent of the well-known American counterpart $BHP = \frac{PLAN}{33000}$)

where N_e = brakehorsepower (metric)

C = constant equal to $\frac{1}{900}$ for four cycle and $\frac{1}{450}$ for two cycle engines

V_h = displacement of a single working cylinder in liters

z = number of working cylinders

n = revolutions per minute or rpm

p_e = brake mean effective pressure or BMEP in kg/cm²

Each of these four variable quantities influences the engine output. But these four variables are not independent of each other, the displacement (V_h) and rpm (n) are particularly closely related. The mean piston speed can be increased only up to a definite limit because of dynamic stresses of piston and crank mechanism and gas pressure losses, so that for a general consideration it may be assumed that

$$c_m = \frac{s.n}{30} \text{ constant}$$

The stroke *s* and cylinder bore *d* in order to produce a conventional form of the combustion space and normal cylinder dimensions can also be varied but little in their relative size, so that it will be assumed here that

$$\frac{s}{d} \backsim \text{constant}$$

The displacement volume of one cylinder

$$v_h = \frac{d^2}{4} \cdot s$$

Therefore the engine output

$$N_e = A_1 \cdot \frac{z.p_e}{n^2} = A_2.d^2.z.p_e$$

where A_1 and A_2 are constant.

Basically nothing will change the sharp decline in cylinder output with rising rpm when simultaneously the mean piston speed is increased 2 to 3 fold and the stroke-bore ratio reduced by 2 to 3. It would decline even with the third power of rpm if it were not for the fact that this well known phenomenon is embodied in the above formula, that under the same conditions the horsepower per liter.

$$N_L = \frac{Ne}{V_h.z} = A_3.n.p_e \quad (A_3 = \text{constant})$$

Guided by the above relations it is possible to answer the question—how must a given engine be modified to secure a desired multiple increase of output. Table 1 gives these changes for the four variables of the output equation and for two additional modifications of cylinder bore for the assumption that the output must be tripled. In addition the basic engine was assumed to have an output of $Ne_o = 1000$ HP, with the mean piston speed and stroke/bore ratio

assumed constant—$c_m = 6\text{m/s}\ \frac{s}{d} = 1.3$;

The other values in the table are self-explanatory.

How the values in the table were arrived at is not going to be further detailed. The four limiting cases, corresponding to the four variables of the output equation are especially noted. It is at once apparent how easy it is to obtain the desired increase in output by increasing the displacement Vh_o, if the increased size of the engine can be accepted. Multiple increase of rpm, n_o, leads to an intolerably high number of cylinders even when the possible increase in mean piston speed—because of smaller cylinder dimensions—would cut the number of cylinders approximately in two. These two lines make it clear why engines of large output in general have cylinders of large dimensions and at the same time are of low rpm, while fast running engines although of small dimensions and high specific output per liter can develop only limited output because of practical limits to the number of cylinders per engine. Even by retaining the original cylinder dimensions ($d_o = 460$ mm $= 18.1''$, $s = 600$ mm $= 23.6''$) and simply increasing their number z_o, a threefold increase in output would lead to an unacceptable $z = 3_{z_o} = 18$ cylinders. On the other hand neither the cylinder dimensions nor their

TABLE 1

CHANGES IN VARIABLE CHARACTERISTICS FOR DIFFERENT WAYS TO INCREASE ENGINE OUTPUT.

N_{e_o} (1000 HP)	$N_e = 3\,N_{e_o}$ (3000 HP)					
d_o	$\sqrt{3}\,d_o$	$\sqrt[3]{3}\,d_o$	d_o	$\frac{d_o}{\sqrt{3}}$	$\frac{d_o}{3}$	d_o
V_{h_o}	$\sqrt{3^3}\,V_{h_o}$	$3\,V_{h_o}$	V_{h_o}	$V_{h_o}/\sqrt{3^3}$	$V_{h_o}/3^3$	V_{h_o}
n_o (300 rpm)	$\frac{n_o}{\sqrt{3}}$	$\frac{n_o}{\sqrt[3]{3}}$	n_o (300)	$\sqrt{3}\,n_o$	$3\,n_o$	n_o (300)
z_o (6)	z_o (6)	$\sqrt[3]{3}\,z_o$ (~ 9)	$3\,z_o$ (18)	$3^2\,z_o$ (54)	$3^3\,z_o$ (162)	z_o (6)
p_{e_o} (5 kg/cm^2)	p_{e_o} (5)	p_{e_o} (5)	p_{e_o} (5)	p_{e_o} (5)	p_{e_o} (5)	$3\,p_{e_o}$ (15)
N_{i_o} (HP/ltr.)	$\frac{N_{i_o}}{\sqrt{3}}$	$\frac{N_{i_o}}{\sqrt[3]{3}}$	N_{i_o}	$\sqrt{3}\,N_{i_o}$	$3\,N_{i_o}$	$3\,N_{i_o}$

Note: The assumed or resulting values of the most important characteristics are given in parenthesis, additional are given in the text.

number need be changed when the brake mean effective pressure (BMEP or p_{eo}) is increased to correspond with the desired increase in output. The relations remain clear and simple and the same increased specific output per liter N_{L_o} is obtained without the excessive number of cylinders.

The above elucidated advantages of increasing the output by increasing the mean effective pressure as opposed to other possible means of doing it have been promoted only in the last 20 years although Buchi began studying them some 40 years ago, and they become the more clear the higher the mean effective pressure is raised. In the last few years a sudden jump ahead achieved results which will be discussed in the following pages. Before that however it must be briefly stated what thermodynamic and mechanical requirements must be fulfilled in an internal combustion engine in order to achieve simultaneously with the increase of mean effective pressure an increased thermal efficiency, i.e., better utilization of fuel as well.

SUPERCHARGING VS. HIGH-PRESSURE CHARGING

In supercharging the working cylinder of an I.C. engine is supplied with more air than it can induct by the outward moving piston out of the ambient atmosphere. A larger charge of air would allow a larger quantity of fuel to be burned therein and a greater amount of energy developed with each filling of the cylinder with consequent higher mean effective pressure and a lower specific fuel consumption.

The power expended to supply the air charge must be kept as low as possible so that the increase in mean effective pressure would not be substantially lowered and the fuel consumption per horsepower of useful load would not be thereby increased unduly. Of the different possible ways to provide the air by a blower the following constitute the principal and limiting possibilities:

a) the blower is driven mechanically by the engine itself, and

b) the energy of engine exhaust gases is utilized in a turbine driving the blower.

The effect of charging is best and clearest shown by the schematic p V diagram—Fig. 1. Because of charging, the compression in the engine cylinder will begin with a higher pressure than atmospheric and a charging loop is formed. It is of positive value however, since the cylinder charging pressure $P_{a,\ zyl}$ is greater than the backpressure p_g. If the charging blower is direct engine-driven the engine exhaust gases are discharged direct into atmosphere with the pressure $p_{g,\ mech}$ whereas for exhaust gas-driven turbocharger the pressure $p_{g,\ abg}$ is higher to account for the pressure drop in the turbine.

The area of the charging loop, which is positive in the supercharged engine, is about 3 to 9% for the mechanically driven blower and 1 to 3% of the total area of the card for the exhaust gas-

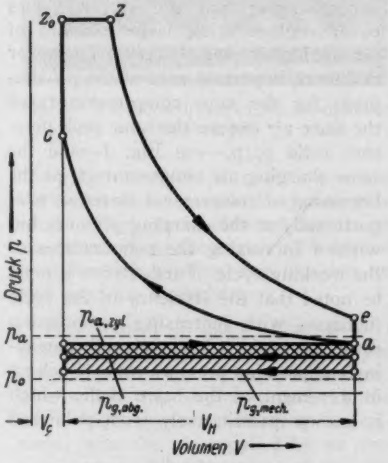


Fig. 1—Schematic pV diagram for supercharging.

p_o —atmospheric pressure
p_a —charging manifold pressure
p_{a-zyl} —charging pressure of cylinder
p_{g-abg} —exhaust pressure in working cylinder when exhausting into turbine
p_{g-mech}—exhaust pressure in working cylinder with mechanically driven blower
V_c —compression space
V_h —displacement volume

driven turbocharger. The power input of the mechanically-driven blower is about double the loop area on the card because of losses in the gear drive, i.e., about 6 to 18%. In the exhaust gas-driven turbocharger however, the total power input to the blower results from additional expansion of gases and the consequent reduction in the charging loop. Thus the mechanical drive results in (6 to 18) less (2 to 6) = (4 to 12)% lesser output and correspondingly increased fuel consumption, as compared with the exhaust gas-driven turbocharger.

The smaller values of these numerical results, obtained experimentally as well as by calculation, refer to smaller en-

gines which operate at full load with smaller air excess and relatively smaller supercharging and the larger values cover engines using large excesses of air and higher supercharging. The upper and more important area of the pV diagram for the same compression ratio, the same air excess, the same peak pressure ratio p_z/p_a—see Fig. 1—and the same charging air temperature t_a at the beginning of compression increases proportionally to the charging pressure but without increasing the temperatures of the working cycle. Furthermore it must be noted that the efficiency of the cycle increases with increasing compression ratio, increasing air excess and increasing compression pressure ratio. Such an improvement of the basic cycle, which is known quantitatively from published research,[1] must actually exist in the engine if its operating conditions are not lower.

The above named influences must all be considered if it is desired to supercharge the engine to get a much higher output and simultaneously or even primarily obtain an improved fuel consumption. Such was the case when MAN decided to continue its development of supercharged diesel engines.[2] A peak pressure of 1700 psig (120 at.) was accepted although it was well known that Dr. Diesel himself failed with pressures at 1278 psig (90 at.) and had to use lower pressures. Meanwhile, of course, over 50 years have passed ond today the newly developed materials and better understanding of the phenomena make such a step possible.

CONSTRUCTION—GENERAL ARRANGEMENT

The construction of an experimental engine which according to preliminary research on normal engines could produce a BMEP of 213-227 psi (15-16 at.) when turbocharged and at the same time have a minimum fuel consumption began several years ago. In addition to the above mentioned basic thermodynamic considerations such questions as stroke bore ratio, number of valves in the cylinder head, trunk piston vs. crosshead, etc., had to be considered in undertaking the design of the new engine. At the same time the new engine had to be suitable for continuous operation, which meant that the stresses in the frame, crank mechanism and especially in the bearings should not exceed the usual limits. Finally a multi-stage type of turbocharger for the high charging pressures—had to be adopted. High pressure charging and the desire for low fuel consumption dictated that the engine could not be below a certain output in order to get a turbocharger of satisfactory efficiency.

All of these considerations resulted in a six-cylinder engine of 11.78″ (300 mm) bore and 17.71″ (450 mm) stroke of crosshead type, Fig. 2. In the crosshead design the piston is free of the increased forces due to angularity of the connecting rod and the separation of cylinder volume from crankcase interior guarantees minimum lubricating oil consumption and is very desirable for heavy oil operation. Bedplate and crankcase were made of welded construction since it surely is quickest and cheapest for a single engine of experimental type. The individual cast cylinders form one continuous block by means of an upper plate and the cylinder interior is separated from the crankcase by cylindrical crossheads sliding in suitably bored out guides.

Tie rods hold together bedplate crankcase and cylinder block—this reliable type of construction is especially suitable for this engine since the combustion peak loads are double the values of a normal type of engine.

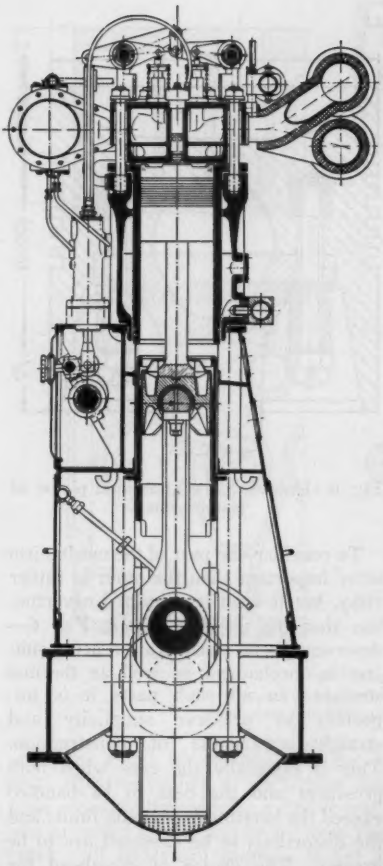


Fig. 2—Cross section of the test engine—MAN type K6V 30/45 bore—11.8″, stroke —17.7″, rpm—400.

The bearings—main, crankpin and crosshead—all are filled with 10% white metal. Despite the doubled combustion peak pressure they are so dimensioned that the normal value of specific pressure is not exceeded so that this simple type of bearing metal can be utilized.

By using two intake and two exhaust valves the valves remain of small size but provide generous flow areas, small losses and an easy arrangement for the

Fig. 3—Welded bedplate of the six-cylinder test engine.

charging manifold on one side and a divided exhaust manifold as normally used in Buchi turbo-charged engines on the other side of the engine with special consideration of flow requirements to and from the turbocharger.

The welded bedplate—Fig. 3—and crankcase were taken over from the original three-cylinder experimental engine which was originally used for the basic high pressure charging experiments with the air supplied by an outside source. All other parts of this engine could be utilized, which was very desirable to accumulation of test experience. This could be done of course only because bedplate and crankcase were of welded type. Fig. 4 is a front view of the engine and shows the charging manifold connecting to the turbocharger on the left, the latter being arranged on the back or exhaust side of the free end of the engine.

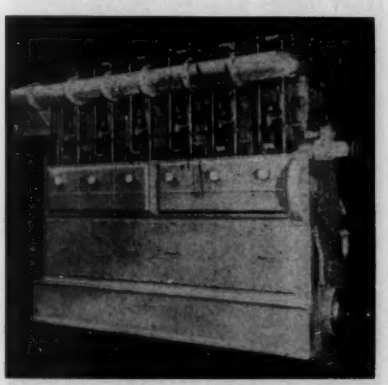


Fig. 4—Front view of test engine—intake side.

DETAILS

The crankshaft—Fig. 5—to meet the doubled peak pressures is of larger size as compared with an atmospheric intake engine of the same cylinder bore. It is however considerably smaller than that of a 50% turbocharged engine of the same cylinder output. The cylinder center to center distance is between the other two.

The chosen type of *crosshead* permits a large bearing surface so that here too the normal specific load could be retained. Reliable bolted joints to the piston rod and to the connecting rod could here be incorporated.

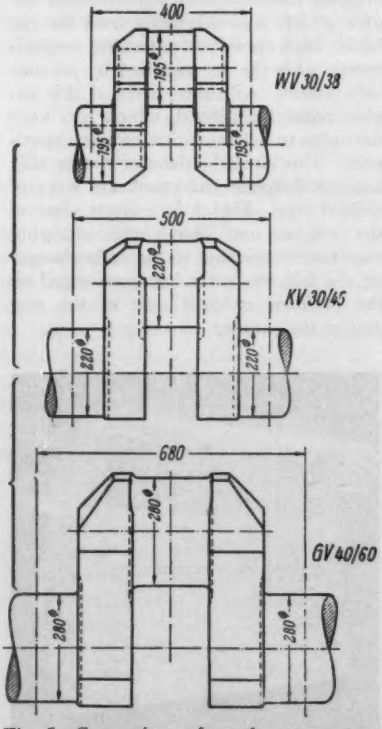


Fig. 5—Comparison of cranks—upper two for the same bore, lower two for the same horsepower.

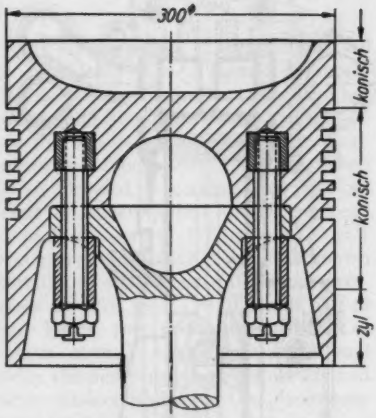


Fig. 6—Uncooled cast aluminum piston of test engine.

To consider one part of the mechanism more important than the other is rather risky, but it must be admitted nevertheless that the *working piston*, Fig. 6—deserves special consideration being subject to mechanical as well as thermal stresses. In all such parts it is important to preserve simplicity and straight-forwardness of construction. This is especially the case when both pressures and the heat to be handled exceed the heretofore common limits and the distortions to be expected are to be greater. The choice of crosshead for the single-acting four-cycle engine is largely in response to the above considerations. Even for a bore of 11.78" (300 mm) the cast aluminum piston could be made uncooled.

The cast iron *cylinder head* is shown in Fig. 7 with a design of one heretofore used on an engine of a similar bore. It differs not only in the number of valves and holding-down bolts but also in shape. The cylindrical outer wall is directly over the cylinder gaskets so that distortions due to high combustion

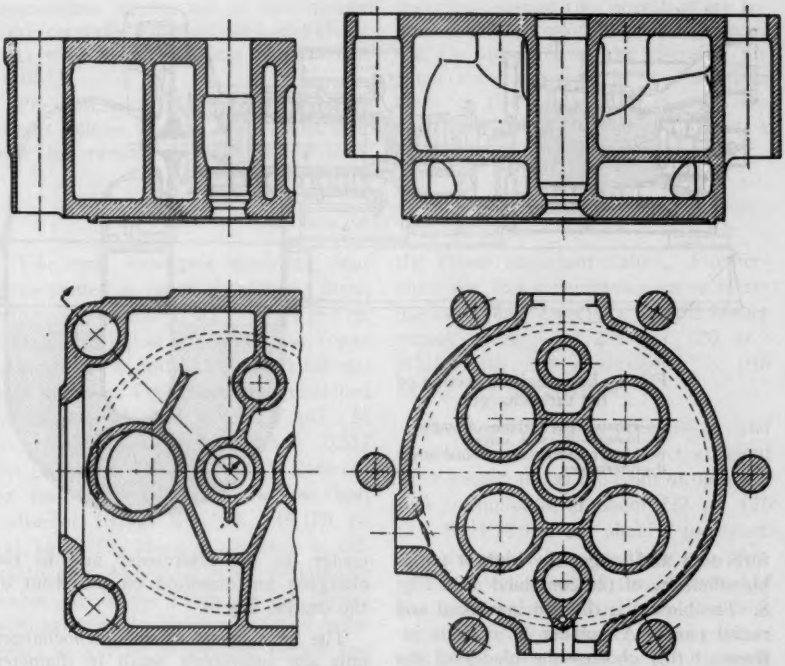


Fig. 7—Cylinder heads.
left—cast iron head of a normal engine of 11.21″ bore
right—cast iron head of test engine of 11.8″ bore

pressures could be avoided. To get a more effective cooling of the bottom flat plate covering the cylinder a double wall was provided as well as ribs between the outer wall and valve cannons to get a forced circulation. The valve guide pistons are provided with rings because of high charging and exhaust pressures and are guarded against side thrust by using rollers on valve levers.

Thus all the details of construction of this new engine are not unusual but only carefully chosen to meet the new working forces and higher thermal loads.

Turbine and blower of the turbocharger are of multi-stage design because both exhaust gases and charging air are of higher pressure than hereto-

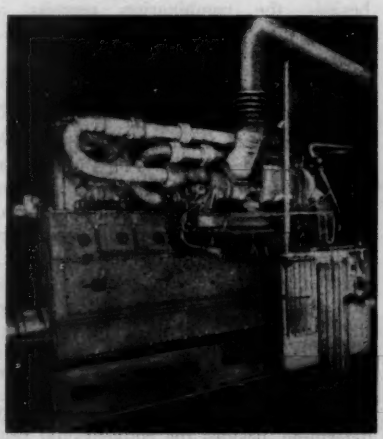


Fig. 9—Rear view of test engine—exhaust side.

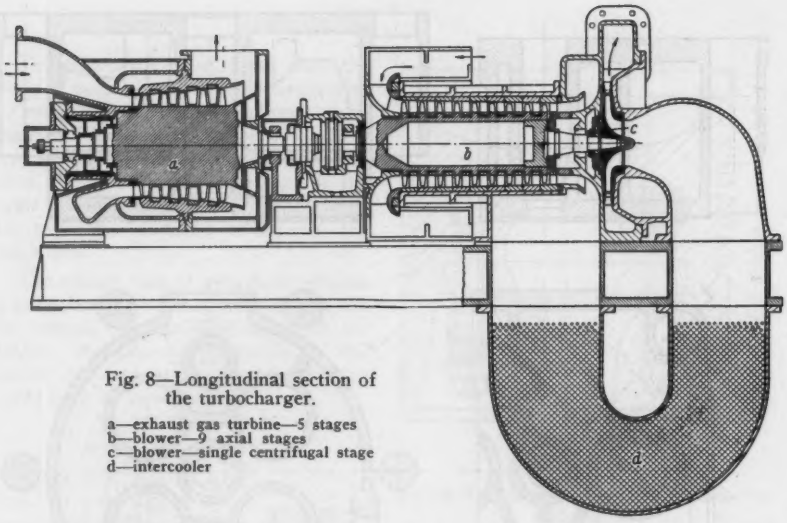


Fig. 8—Longitudinal section of the turbocharger.

a—exhaust gas turbine—5 stages
b—blower—9 axial stages
c—blower—single centrifugal stage
d—intercooler

fore used and to get the highest possible efficiency of the combined unit, Fig. 8. The blower is divided into axial and radial parts. A number of reasons influenced this choice—the blades of the higher stages of the axial blower for this size of engine would be so short that its efficiency would be jeopardized, besides the combination permits a smooth flow of air to and from intercooler, to the aftercooler and to the charging air manifold on the front of the engine, Fig. 4.

The axial parts of the turbocharger unit are sufficiently small in diameter that it could be placed quite close to the engine and the centrifugal blower with its diffuser had ample room on the end of the engine, Fig. 9. The turbocharger normally runs at about 12,500 rpm.

TEST RESULTS—CONDUCT OF TESTS

Both authors went to great pains to secure reliable results. This does not mean however that the engine was in any way babied or specially set up. Quite the contrary, we endeavored to evaluate the condition of the engine, after many hundreds of hours of operation and made a conscious effort to see that the measurements made for the steady-state condition were adequate in extent, accuracy and painstaking care.[8]

It was also necessary of course to determine, besides the conventional test data, such values which would clearly indicate the inner phenomena and their influence on the question whether the engine could satisfactorily carry the increased loads indefinitely. In addition to indicator cards over 80 values were recorded for each observation point. It goes without saying that all instruments and measuring devices were carefully calibrated and continuously tested to assure perfect operation. Control calibrations and fuel analysis were also carried out by the Technical University München and at the Physical-Technical Federal Laboratory in Braunschweig.

Instructions on the use of instruments were carefully complied with and check tests were made to assure reproducible results.

The evaluation of the most important results follows. Due to space limitations only the results of the series of tests covering constant rpm operation are reproduced. Operation at propeller speed and the influence of the charging air temperature can only be mentioned in the hope that opportunity may present itself to publish further details at a later date.

GENERAL OPERATING DATA

The most important operating data were plotted as functions of brake mean effective pressure (BMEP = p_e) in Fig. 10.[4] A BMEP of 213 psi (15 at.) was assumed as a normal full load for design purposes. Fuel consumption reached its lowest value at about 227 psi (16 at.) and was definitely below 0.313 lbs./BHP-hr (140 g/PSeh.* Referred to fuel used, which had a low heat valve of 18,300 BTU/lb (10,170 K. cal/kg). The thermal efficiency is calculated as 44.6%—a most remarkable value. It is the result of the proper utilization of the above mentioned influences as well as of reduced friction loss to be discussed later.

Many check measurements are entered in Fig. 10. Arrows, indicating the lowest range, confirmed again and again the above important values. Furthermore the fuel consumption curve is remarkably flat, especially at still higher output ratings—at 284 psi (20 at.) BMEP the fuel consumption is still barely .317 lbs/BHP-hr.

The supercharging pressure p_a and the back pressure p_g intersect at about ⅓ of output and at the point of optimum fuel consumption at about 227 psi (16 at.) BMEP they are about 7 psi apart. The difference between these two pressures is indicative of the efficiency of the turbocharger. This very favorable value results for instance in the exhaust gas temperature before the turbine being only 932° F. (500° C.). It rises then with the rising temperatures of charging air to about 1040° F. (560° C.) and stays thereafter still below 1112° F.

**Translator's Note: In Germany the low heat value of fuel is used in calculating the efficiency of an internal combustion engine. In the United States the high heat value is in common use and is specified by the ASME Power Test Code. The difference between the high and low heat values is the heat loss due to hydrogen content of fuel yielding water as one of the products of combustion, which amounts to about 5½%, but can be determined accurately only when the chemical composition of fuel (hydrogen content) is known. The fuel consumption figures are obviously not at all affected by the heat value of the fuel but the German figures for thermal efficiency are somewhat greater because they are referred to low heat value. However, German fuel consumption figures refer to the metric horsepower PS_e which is 98.63% of BHP. Furthermore, while the lubricating oil pump of the engine tested was driven by the engine the circulating water pump was driven by an electric motor but the power required by this motor (about 3 KW at full power of engine) was not deducted from the useful output of the test engine—a customary German procedure.*

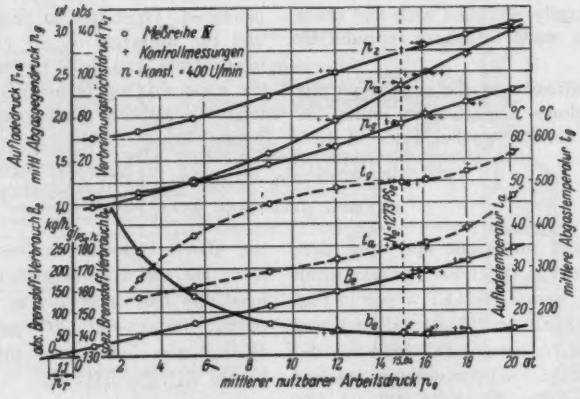


Fig. 10—Values for operation at constant rpm (n = 400) with different mean effective pressures (BMEP).

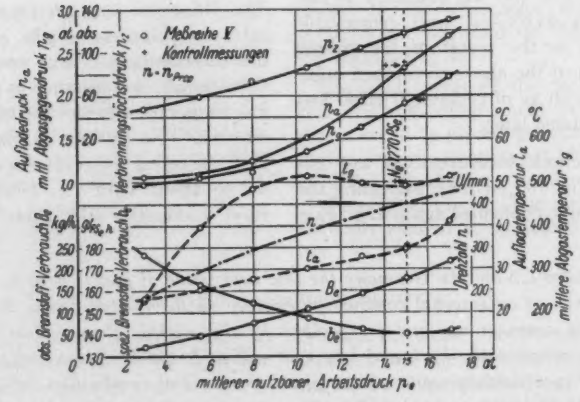


Fig. 11—Values for operation at propeller rpm.

Figs. 10 and 11.

p_a—charging pressure
p_g—mean exhaust back pressure
p_z—peak combustion pressure ata-atmospheres absolute (14.2 psi)
o —Messreihe IV-Series IV
\+ —check measurements
n —Drehzahl = rpm
B_e—Brennstoff-Verbranch kg/h = total fuel consumption
b_e = specific fuel consumption g/PSeh
p_r—mean friction pressure at idling
p_e—mittlerer nutzbarer Arbeitsdruck = BMEP
t_g—mean exhaust temperature in °C.
t_a—charging air temperature

(600° C.), a value reached with conventional turbocharger for half as large BMEP. This temperature of 1112° F. (600° C.) is still within the limit of temperature for continuous operation of an uncooled turbine impeller.

The right branch of the charging air temperature curve t_a is somewhat steeper because the charging air cooler was designed for a BMEP = 213 psi (15 at.) and for higher loads could not cool it any lower—especially on test days with high ambient air temperatures. From the similar curvatures of the charging air curve t_a and of the exhaust temperature curve t_g their interdependence is plainly discernible. The cooling of the charging air is therefore essential not only for the more efficient filling of the cylinder but also for the possible increase of the BMEP carried by it. Besides the investigation of the influence of charging air temperature has shown that fuel consumption rises with the rise in air temperature. At full power, air temperature rise from 93.5° F. (34.1° C.) to 158° F. (69.8° C.) resulted in .0053 lb/BHP-hr (2.4g/PS_{eh}) increase in fuel consumption, which corresponds to an increase of 1.7%.

Finally, the combustion peak pressure is to be noted, for a BMEP of 213 psi (15 at.) it reached a maximum of 1700 psig (120 at.) as anticipated, while the compression pressure rose to 1164 psig (82 at.) due to the retained high value of compression ratio of E = 13. The peak pressure ratio $p_z/p_a = 120/2.35 = 51$ is of normal value for engines of large output and of conventional type despite the high combustion pressure. The retention of this ratio for increased compression and combustion pressures is, as already shown, very important for attaining low fuel consumption and low exhaust temperature.

Of interest are the differently varying values of these characteristics when propeller load is carried by the engine, Fig. 11. In the common point, corresponding to 400 rpm and 213 psi (15 at.) BMEP, it is obvious that the same values will exist in both diagrams. The pressures and temperatures before as well as after the turbocharger and also the fuel consumption will be different however due to lower rpm and correspondingly higher BMEP for the same output. The slight hump in the exhaust temperature curve t_g is to be noted. This results from the fact that, according to propeller law, the engine output is proportional to the third power of rpm whereas the BMEP varies as the square of rpm. Other influences are the charging air temperature and the operational characteristics of the turbocharger. However, the exhaust temperature at this point is still low at 960° F. (515° C.).

MEAN WORKING AND FRICTION PRESSURES

The combustion peak pressures and thereby the combustion phenomena in all cylinders were continuously observed by means of pulled diagrams from a conventional mechanical indicator. A DVL glow lamp indicator of the DVL (Deutsche Versuchsanstalt für Luftfahrt) balanced diaphragm type was installed on cylinder number 1 for continuous checking of high pressures within, for determinations of mean indicated pressures and of friction pressures.

On Fig. 12 are shown three representative indicator cards, corresponding to full power of the engine when (1) not supercharged, (2) conventionally turbocharged and (3) high-pressure turbocharged. The beginning of the combustion and its progress show a normal picture of a well-tuned diesel engine. The combustion pressure rise, except for very small loads, shows 28-42 psi. (2-3 at.) and at overload about 52 psi (3.7 at.) per degree of crank

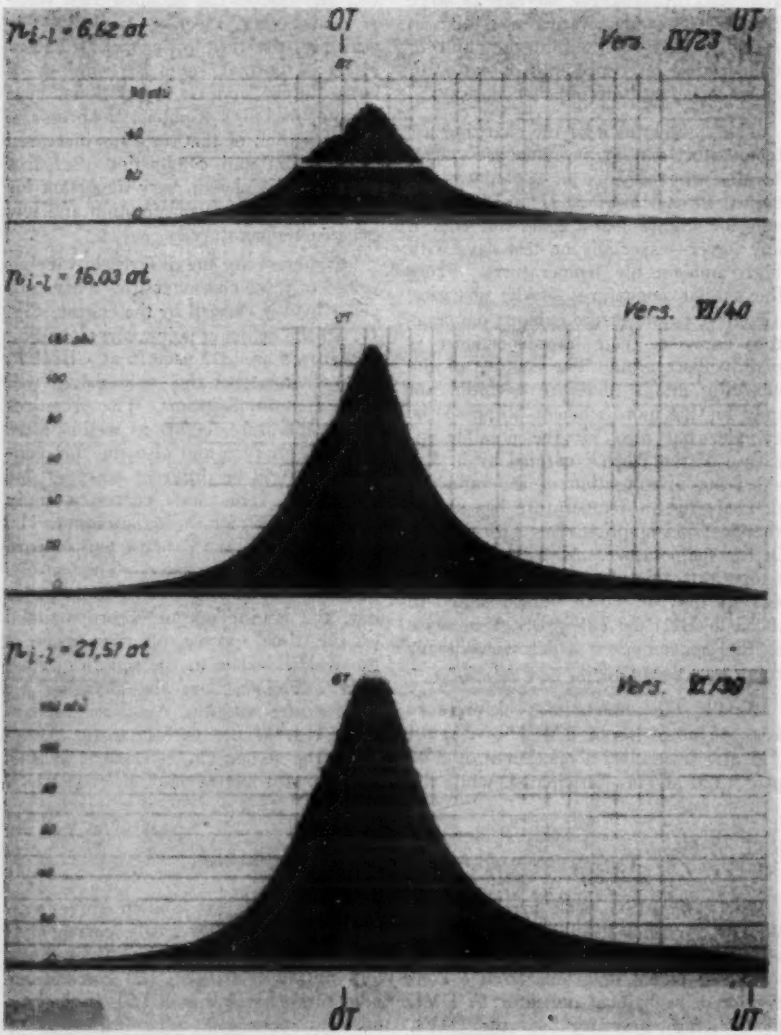


Fig. 12—Pressure-time indicator cards for three different outputs at 400 rpm. OT—top dead center, UT—bottom dead center p_{i-l} = mean indicated pressure *without* crediting pressure loop.

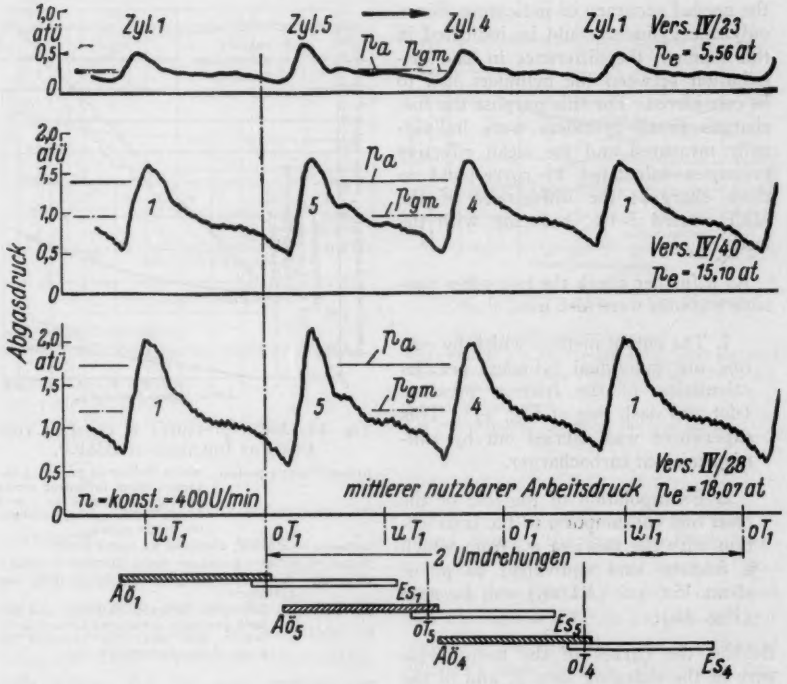


Fig. 13—Exhaust back pressure between engine and turbine for three different outputs at 400 rpm.

p_a —charging pressure
p_{gm}—mean exhaust back pressure
p_e —BMEP
OT—top dead center
UT—bottom dead center
2 —Umdrehungen = 2 revolutions
Abgasdruck atü—exhaust pressure—atmospheres gage

movement corresponding to perfectly normal value of a quiet burning. The "weak spring" cards are not reproduced here for lack of space. Reproductions of exhaust gas back pressure oscilograms for three different loads are added in Fig. 13. Even though the exhaust gas pressure peaks rise to about 5.4 psig (18 at.) there is available a substantial scavenging air pressure drop to be made effective through valve overlap in the upper dead center. The more the back pressure drops the easier it is to scavenge the gases out of the cylinder, since not only is the available scavenging air drop greater but the residue of combustion gases gets less at the same time.

The *friction loss*, Fig. 14, is of special interest with so high a BMEP. The difficulty of measuring it lies in the fact that the difference between two large values of the mean indicated pressure p_i and of the mean effective pressure p_e is small so that a high accuracy of measurement must be required. Check measurements with different membranes in the DVL glow lamp indicator and elimination of all possible errors in the marking of the dead centers secured

the needed accuracy of indicating. Since only one cylinder could be indicated in this manner, the difference in load distribution between the cylinders had to be considered. For this purpose the fuel charges to all cylinders were individually measured and the mean effective pressures calculated to correspond to these charges; the differences of the MEP varied 5-3%, reducing with the load.

As a further check the two other possible methods were also used.

1. The cutout method which by cutting out individual cylinders permits calculation of the friction pressure (dot and dash line of Fig. 14). This experiment was carried out by *running without* turbocharger.

2. Extrapolation of the line of the total fuel consumption to the intersection with the abscissa p_e, from which a friction load equivalent to p_r of about 15.6 psi (1.1 at.) can be read (Fig. 10).

Besides the curves of the mean pressure of the charging loop p_e and of the mean charging pressure p_a and the mean back pressure p_g, i.e., immediately before and after the working cylinder, are rationally spaced with the difference between the latter increasing with load and consequent weight of gas.

These checks first of all show adequately that the engine possesses very favorable friction characteristics as a result of its construction and load distribution, with the friction loss being

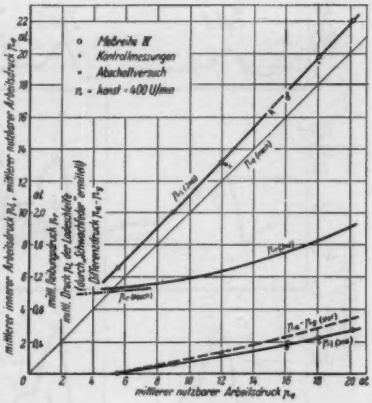


Fig. 14—Mean pressures at constant rpm (400) as functions of BMEP.

$p_{i(Ind)} = p_{i1} + p_{e(Ind)}$ mean indicated pressure obtained from indicator cards of high pressures p_{i-1} and from "weak spring" indicator cards $p_{i(Ind)}$

$p_{e(mech)}$ = BMEP, obtained by water brake

$p_{r(Ind)} = p_{i(Ind)} - p_{e(mech}$—mean friction pressure

$p_{r(abcch)}$—mean friction pressure by cutting out cylinders

$p_a - p_{g(stat)}$—difference between charging and gas back pressure measured immediately before and after the cylinder by damped measuring devices

relatively very small for the high BMEP of the engine.

These experiments have shown that the friction loss p_r rises with increasing load. While heretofore it was customary to accept the friction load as constant at constant rpm and within the usual load range from no load to full load (64-85 psi BMEP), it is no longer possible to do so for the large load increases, as was indicated above.

EFFICIENCIES AND EXCESS AIR

The difference between the total efficiency η_e of an actual engine and the efficiency η_v of the comparison cycle of a perfect engine, Fig. 15, results from losses in the actual engine. These are due to flow or combustion, thermal or mechanical. When the operating conditions raise the efficiency of the comparison cycle this in itself will have a favorable influence on the actual efficiency. Next in importance to the already mentioned compression ratio and combustion pressure ratio is the excess air ratio which expresses the amount of air in

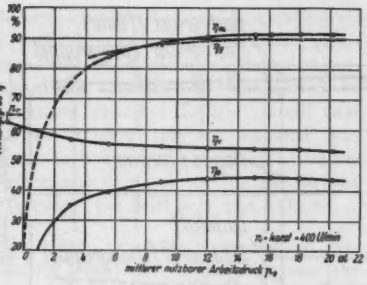


Fig. 15—Engine efficiencies.

η_e —thermal efficiency
η_T —efficiency of the perfect engine of comparison cycle
η_m—mechanical efficiency
η_g —perfection coefficient
p_r —mean friction pressure at idling

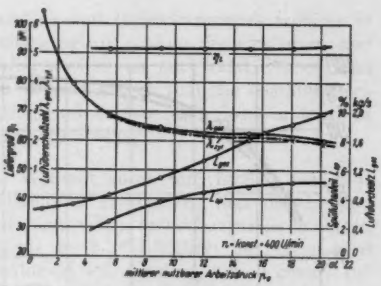


Fig. 16—Air ratios of the engine.

η_v —filling coefficient for cylinder
λ_{ges} —excess air ratio referred to total air used
λ_{zyl} —excess air ratio referred to air left in cylinder
L_{ges}—total air used
L_{sp} —scavenging portion of the total air used

excess of theoretical combustion requirement.

The greater the excess of air the more favorable become the thermodynamic values because of reduction of CO_2 and H_2O content in the combustion gases; for the same reason the efficiency η_v rises with dropping load—independently of the influence of the combustion pressure ratio. The limiting value, with $p_1 = 0$ and corresponding infinitely large excess of air, i.e., pure air, is equal to the efficiency of Carnot cycle for a given compression ratio. To the total air furnished by the turbocharger L_{ges} corresponds an excess air coefficient λ_{ges} which first reduces quickly with rising load, Fig. 16, but later the influence of turbocharging becomes felt and throughout a load range from 170 psi to 284 psi of BMEP (12-20 at.) it stays only very little above 2, i.e., twice the theoretically required amount of air.

The measured fuel consumption referred to the low heat value of the fuel oil used give the *thermal efficiency* η_e, which at full load is about 45% in round figures, the first time such a remarkably high value was reached. Because of low friction losses on one hand and the high BMEP of 213 psi (15 at.) and over on the other the mechanical efficiency rose to over 91%. The flatness of the curves of both efficiencies over a wide range of load is worth noting.

Out of the efficiencies of the basic cycle, mechanical and thermal efficiencies is obtained the *perfection coefficient* $\eta_g = \eta_e/\eta_v \cdot \eta_m$, which represents a measure of losses through pressure and temperature changes during charging and exhaust strokes, of imperfection of combustion and of losses of heat during the working cycle. The curve of its values throughout the load range is also very flat and with 90% in round figures shows a remarkable state of perfection observable in very good combustion and in little heating up of cooling water. The exhaust gases remained completely invisible even under highest loads.

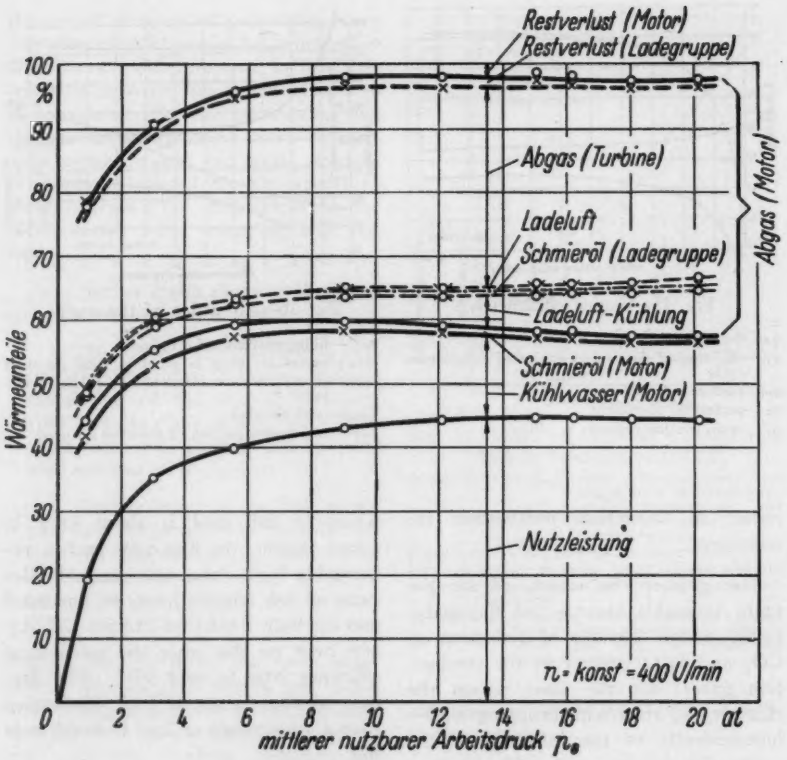


Fig. 17—Heat balance for the engine tested.

Restverlust (Motor)—residual loss (engine)
Restverlust (Ladegruppe)—residual loss (turbocharger)
Abgas (Turbine)—exhaust gas (from turbine)
Ladeluft—charging air
Schmieröl (Ladegruppe)—lubricating oil (turbocharger)
Ladeluft-Kühlung—charging air coolers
Schmieröl (Motor)—lubricating oil (engine)
Kühlwasser (Motor)—circulating water (engine)
Nutzleistung—useful output
Abgas (Motor)—exhaust gas (from engine)
Wärmeanteile—percent of heat

HEAT BALANCE

The heat rejected to the circulating water is taken from the *heat balance,* Fig. 17, which gives a further chance to visualize the newer relationships of the engine. Since an engine employing an exhaust gas driven turbocharger and cooling of charging air is being considered it is necessary to differentiate whether the engine alone or the combined unit with turbocharger is to be the basic unit.

In the first case the heat of the exhaust gases is measured immediately after the exhaust valves. In addition to the usefully employed heat, the losses to

the circulating water and lubricating oil as well as that in the exhaust gases there will be a remainder due to unmeasurable radiation and inaccuracy of measuring devices. In the second case the subdivision due to exhaust gas driven turbocharger goes deeper and the temperature of gases leaving the turbine determines the heat lost into exhaust.

In either case the heat lost to circulating water is the same because the turbine is uncooled. As indicated in the discussion of the perfection coefficient the heat lost to water is estimated to be 12% at the optimum load—a value never before reached in internal combustion engines. In the lubricating oil from the engine and turbocharger as well as in charging air cooling through inter- and after-coolers an additional 8% of heat is carried off, so that the combined engine and turbine unit loses a total of 20% to cooling.

Without going into further details it should be emphasized that the division of heat when turbocharger is included leads to 1-2% higher, but of a total value of 4% residual, which covers the additional losses of the uncooled gas turbine including piping between temperature measuring stations. That this circumstance is so clearly measurable is further proof of the accuracy of the tests.

HEAT STRESSES

Even though the heat carried off by the circulating water of the engine is a measure of the mean heat load of the walls around the combustion space, nevertheless some temperature differences in selected positions may be of much interest. Thus the temperatures in the walls must be measured. In this connection a comparison of piston wall temperatures of two different engines shown in Fig. 18 is of interest. The points where the temperatures were measured are also shown on Fig. 18. The influence of the material of the pistons—cast iron vs. aluminum, for the normally supercharged engine is also shown. Comparable are only the curves for the same material (aluminum) c and d with e and f. The rise of temperature with increasing load is slower for the new high-pressure charged engine and even for a BMEP = 255 psi (18 at.) reaches a tolerable value of 572° F. (300° C.).

In other experiments the temperatures of piston, cylinder head and cylinder liner were measured. In the center of the piston the measuring point was .150″ (3.8 mm) from the surface, the location of others is shown on Fig. 19. Here the piston temperatures are somewhat higher than in Fig. 18 due to the location of measuring points being nearer to the piston top, otherwise the previous measurements were confirmed. Obviously the highest piston temperatures are in the piston top. It is important to note, however, that the temperature at the first ring is already considerably lower, which naturally lightens its service requirements. The temperature measurements were made with an iron-constantan thermocouple suitably compensated. A periodic contact near the lower dead center region served to carry the current.

The *heat flow* from gas to cooling water increases with the load in the high pressure charged engine also. It can be calculated beforehand and permits comparison with measured values. The heat flow depends on gas temperature changes and the heat transfer conditions. The gas temperatures during the working cycle for constant compression and combustion conditions are a

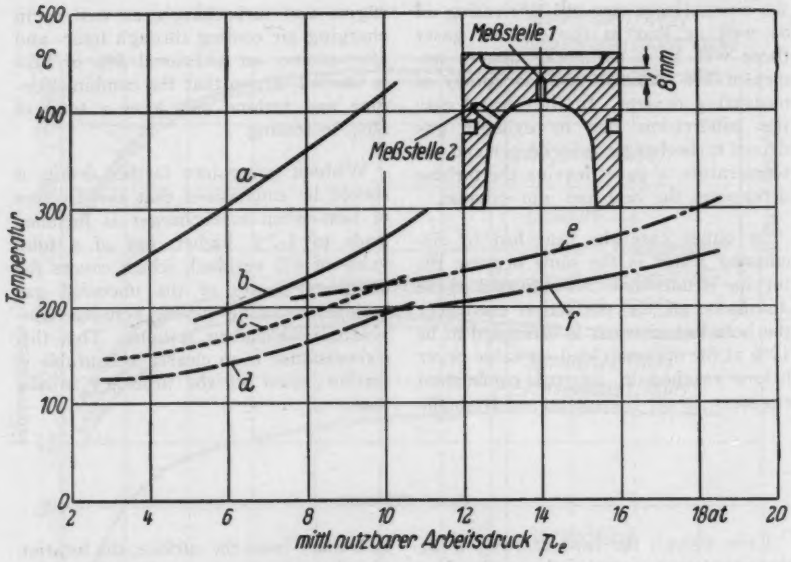


Fig. 18—Comparison of piston temperatures.

Engine type GV 28.5/42 of 11.22″ bore, 16.54″ stroke Messstelle = measuring point.
Test engine KV 30/45 of 11.8″ bore, 17.7″ stroke.

Curve	*Type*	*Piston Material*	*Measuring Point*
a	GV 28.5/42	cast iron	1
b	GV 28.5/42	cast iron	2
c	GV 28.5/42	aluminum	1
d	GV 28.5/42	aluminum	2
e	KV 30/45	aluminum	1
f	KV 30/45	aluminum	2

function of the charging air temperature t_a and of the air excess rate λ_{zyl} of combustion. Because of the effective inter- and after-cooling the charging air temperature rises but little. Throughout the range of load from idling to full load (BMEP = 284 psi or $p_e = 20$ at.) from 72 to 115° F. (22 to 46° C.) or only about 8% (see Fig. 10) in absolute temperature which alone affects the cylinder filling coefficient (see Fig. 16). Because the air excess ratio under higher loads drops but very slowly (see Fig. 16) with the charging process becoming more efficient, the temperatures of the working cycle through this load range increase very little. Actually the indicator cards beginning with BMEP of 170 psi ($p_e = 12$ at.) to the maximum give almost the same temperature curve and thereby similar mean temperatures t_{gm}.

If the heat transfer coefficient between combustion gas and the gas-contacted surface of the combustion space would remain constant then this mean temperature t_{gm} would determine the heat loss. However, in the range of high gas temperatures—within the indicator card and simultaneous high gas pressures the heat transfer is roughly ten times better than with low temperatures. For the

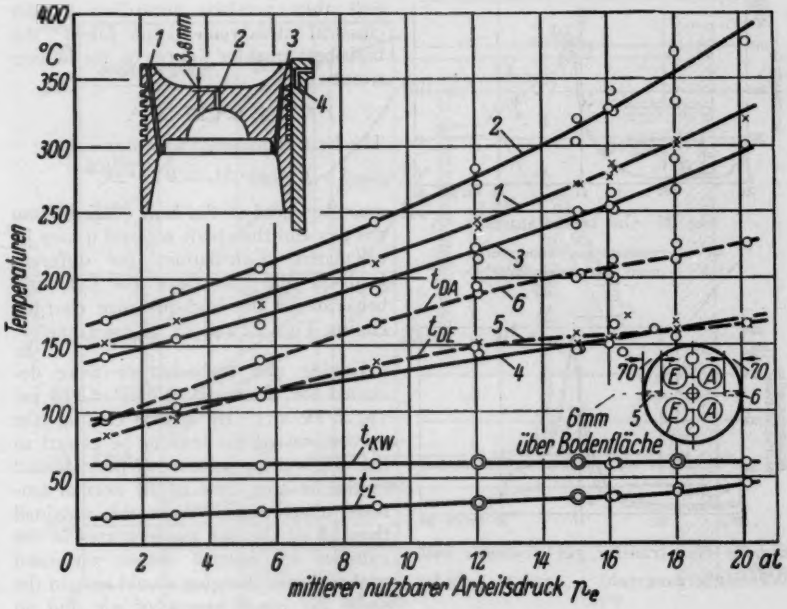


Fig. 19—Wall temperatures of combustion space.
über Bodenfläche—over bottom surface.
The curve figures correspond to measuring points given on sketches.

E —intake valves A—exhaust valves t_{DE} —wall temperature cyl. head bottom, intake side
t_{KW}—water discharge temperature t_{DA} —the same, exhaust side
t_L —charging air temperature

heat transfer measurements so far available on four- and two-cycle engines the following formula has proven useful—$\alpha_g = k\sqrt{pt}$, where the coefficient k depends upon the turbulence within the cylinder and may be assumed[5] as $k = 2.1\sqrt[3]{c_m}$ where p is in kg/cm², T is absolute temperature in °K, c_m is the mean piston speed in m/s.

The resultant mean gas temperature used in heat transfer then follows from a simple consideration of balance[6]

$$t_{g,\,res} = \frac{(\alpha_g\, t_g)m}{\alpha_{gm}}$$

The variation of t_g and α_g and determination of $t_{g,\,res}$ is shown in Figs. 20 and 21 for several indicator cards. For comparison one full load card of an unsupercharged engine is also shown.

The *heat flow density* q (in Kcal/m²L) through a flat wall of the combustion space, for instance through the cylinder head bottom wall ($\delta = 25$ mm, $\lambda_{Guss} = 45$ Kcal/mh° C.), could be easily calculated given the mean cooling water temperature t_{KW}.

$$q = k\ (t_{g,\,res} - t_{KW})$$

when the heat transfer coefficient δ_{KW}

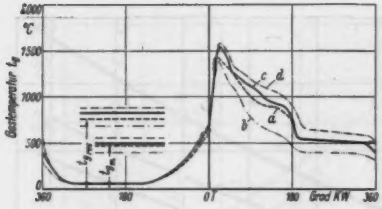


Fig. 20—Gas temperatures.

$t_{g,\,res}$—resultant gas temperature
t_{gm} —mean gas temperature

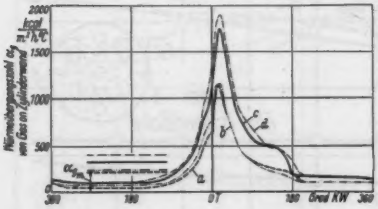


Fig. 21—Heat transfer, gas to cylinder wall.
Wärmeübergangszahl α_g, gas to cylinder wall.

Figs. 20 and 21—Determination of resultant gas temperature $t_{g,\,res}$ of the working cycle.

a BMEP = 80.6 psi (unsupercharged, full power)
b BMEP = 79.0 psi (supercharged)
c BMEP = 228.0 psi (supercharged)
d BMEP = 286.0 psi (supercharged)
OT = top dead center; Grad KW = degrees of crank angle

on the water side were known, since

$$\frac{1}{K} = \frac{1}{\alpha_{gm}} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{kw}}$$

However for high heat loads q, such as are under consideration, the boiling temperature in the boundary layer on the water side must be exceeded, because without boiling the needed high heat transfer could not exist in the cylinder head with the low water velocities. (See the temperature diagram of Fig. 22 for proof.).

When the boiling temperature be exceeded by Δt_s in the boundary layer, it will then produce according to the classical measurements of Jakob[7] the high heat transfer figures on the boiling water

$$\alpha_s = 0.21 \cdot \Delta t_s^4.$$

The heat transferred to water

$$q = \alpha_s \cdot \Delta t_s = 0.21 \cdot \Delta t_s^5$$

must be equal to the heat coming from the gas and thus both Δt_s and q may be calculated. Calculations for different loads at 400 rpm show the following behavior of the high-pressure charged engine, Figs. 22 and 23.

Engine and turbocharger were designed for a normal BMEP = 213 psi (p_e = 15 at.). By heavy cooling the precompressed air was to be cooled to the original temperature if possible and in the limiting case to the normal ambient temperature. Were this obtained then all of the gas temperatures in the cylinder for normal output with and without supercharging would remain the same for equal excess of air and so would their mean values as well as the resultant gas temperatures $t_{g,\,res}$ which would stabilize at about 540° F. (300° C.) higher.

Actually with supercharging to 2.35 times the outside pressure all of the pressures are raised in this ratio. Since α_g varies approximately proportional to $\sqrt{P}$, for normal load of supercharged engine α_g will be $\sqrt{2.35} = 1.53$ times higher and the heat flow density should increase in the same ratio.

However, since cooling can be carried out successfully only down to 94° F. (34.5° C.) i.e., to ratio $T_L/T_{LO} = 1.07$, so that $T_{g,\,res}/T_{g,\,res,\,o} = 1.07$ the value of q will also increase as a consequence.

The hourly loss to cooling water Q_{KWMOT} (in kcal/h) should be very nearly proportional to the heat flow q (in kcal/m²L) calculated from the output, so that it should increase similarly.

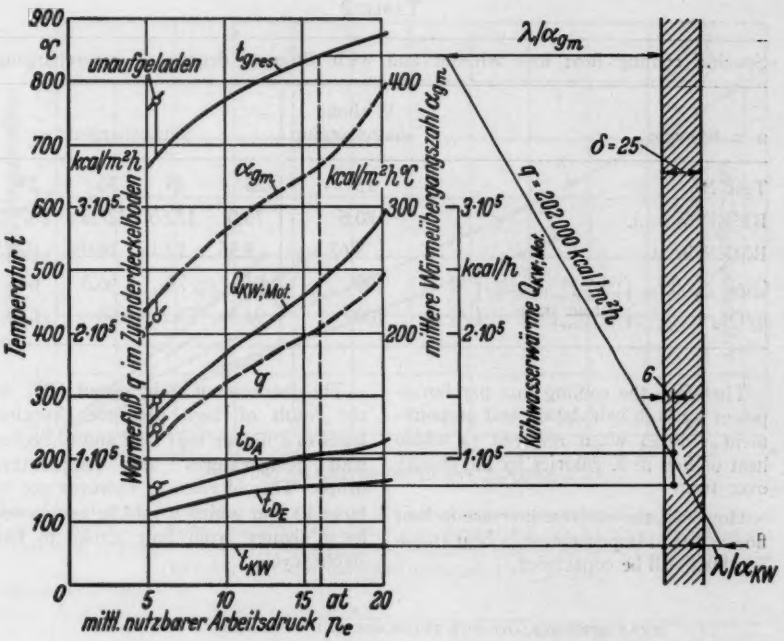


Fig. 22—Heat transfer through the walls of combustion space.

Fig. 23—Temperature diagram for the cylinder head bottom for a load of 287 psi BMEP.

$Q_{KW\ Mot}$—measured heat of engine cooling water
q —calculated heat flow in cylinder head bottom
α_{gm} —mean heat transfer coefficient gas to wall during a single cycle (see figs. 20 and 21)
$t_{g,\ res}$ —resultant gas temperature (see figs. 20 and 21)
t_{KW} —cooling water temperature
t_{DE} —wall temp. of cylinder head bottom—intake side
t_{DA} —the same—exhaust side
unaufgeladen = unsupercharged
mittl. nutzbarer Arbeitsdruck p_e = BMEP

λ = 45 kcal/mh°C heat transfer coefficient for cylinder head bottom (cast iron)
q = 202000 kcal/m²L heat flow through cylinder head bottom
α_{KW} = heat transfer coefficient on the water side

The corresponding curves of Fig. 22 show a good compliance with this requirement. From the measurements of cooling water the specific cooling heat loss referred to heat energy of the fuel Q_{Br} is calculated. It shows a definite decrease in percent with increasing supercharging which corresponds very well with the values calculated from q, (Table 2).

	normal output, unsupercharged BMEP = 79.6 p.s.i. (5.6 at)	to	normal output, supercharged BMEP = 213 p.s.i. (15 at)
FROM	100%	to	67% or ⅔

TABLE 2

Specific cooling heat loss without and with different degree of supercharging

n = 400 rpm	Without Supercharging	Supercharged			
Test No.	15	23	25	36	39
BMEP in p.s.i.	80.6	79.0	172.0	227.5	287
BMEP in at	5.67	5.56	12.1	16.05	20.15
$Q_{KW\ Mot}/Q_{Br}$ {refer to BMEP = 5.67 at. without supercharging} %	100	94	74	66.3	64
q/Q_{Br} {refer to BMEP = 5.67 at. without supercharging} %	100	94.5	76.2	66	63

Therefore the cooling loss per horsepower by both calculation and measurement reduces when referred to added heat of fuel in K calories by the change over from

However, the *absolute* increase in heat flow q due to improvement in heat transfer must still be considered.

The increase of q by about 60% as the result of high pressure charging leads to a higher heat load and to higher wall temperatures and temperature drops. This increase is however not as large as that which would be occasioned by a change from four stroke to two stroke cycle.

HEAT STRESSES, OUTPUT INCREASE AND DIMENSIONS COMPARED

The above mentioned relations lead to very interesting results of comparison with other supercharged engines. For this purpose former results are reviewed[8] and correspondingly modified, Figs. 24 and 25.

The *increase in heat carried off by circulating water* as a function of the mean indicated pressure for different supercharged engines is shown on Fig. 24. Engines with conventional turbocharging show the steepest increase. The new engine has a smaller rise although not as small as the experimental engines of Smith (Ricardo) and Büchi, which are however not immediately comparable because they were supercharged with foreign air and therefore had no back pressure and had correspondingly lower exhaust temperatures. The higher values of air ratio for turbocharging in Fig. 24 refer to point of origin designated by p_{i_o} which despite the different absolute values for different engines falls into value of ratio equal to one; with increasing output the air ratio becomes smaller.

In engines with a larger air ratio the heat in cooling water becomes less, which is quite natural when following the above analysis. It is however not obvious that such substantial output increases are obtainable as were demonstrated by the new engine; the reasons for it have been discussed above.

For the engines represented on Fig. 24, Fig. 25 gives the relation between mean indicated pressure p_i and the charging air pressure p_a. The steep yet bent curves of the WV and MV types of MAN engines are characteristic of the heretofore built turbocharged engines. They reach high MIP's (p_i) with low charging air pressure p_a; these curves break off soon however because the exhaust gas temperature

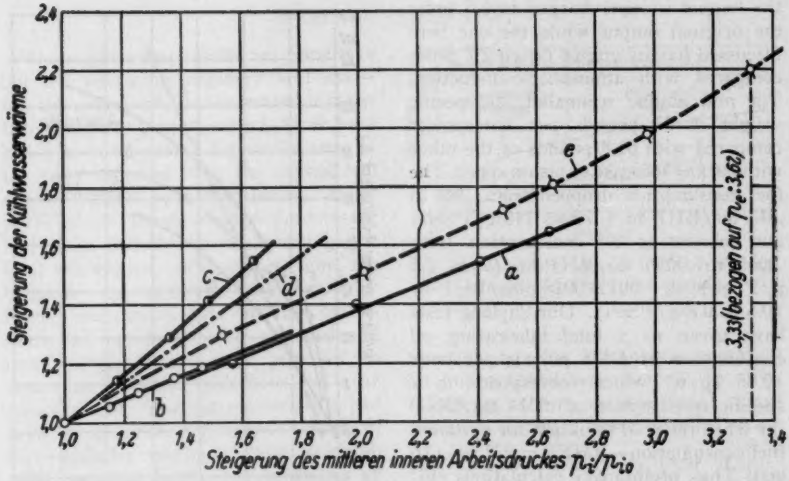


Fig. 24—Increase in cooling water heat in different supercharged diesel engines as function of increase in mean indicated pressure —p_{io}. Steigerung der Kühlwasser-wärme = increase in cooling water heat. Steigerung des mittleren inneren Arbeitsdruckes p_i/p_{io} = increase in mean indicated pressure ratio p_i/p_{io}.

bezogen auf = referred to

Curve	Year	Type	Bore-Stroke ins	RPM	Initial MIP or pio		Type of Supercharging	Excess Air Ratio	Compression Ratio
					ats	psi			
a	1911/14	Buchi	8.66 • 13.8	300	5.7	81.0	foreign	1.7	—
b	1936/38	Smith (Ricardo)	4.5 • 5.0	1500	14.2	202.0	foreign	1.4	1:10
c	1936	MAN-WV	11.8 • 15.0	700	7.3	103.7	turbo	2.14 to 1.3	1:12.6
d	1942	MAN-MV	15.7 • 18.1	450	6.95	98.7	turbo	2.5 to 1.6	1:13
e	1951	MAN-KV	11.8 • 17.7	400	6.62	94.0	turbo	2.8 to 1.9	1:13

limit combined with a small excess air ratio (see figures given under Fig. 24) are quickly reached.

The original experiment with foreign charging air supplied to the engine in which the blower horsepower did not have to be accounted for demonstrated despite the air ratio being kept constant and by comparable quality of combustion a maximum BMEP of about 240 psi (17 at.); this would correspond under optimum conditions to about 200 psi (14 at.) for the exhaust gas turbocharged engine.

For the new turbocharged engine, designated as type MAN-KV, the mean indicated pressure p_i as function of charging pressure, p_a rises practically as a straight line up to the last measured BMEP of 312 psi (22 at.). Our tests have shown that at that point neither exhaust gas temperature, nor the air ratio or the other stresses at this maximum output have reached their limit. The *outside dimensions and the specific weight* are two other practically important characteristics. Table 3 shows two engines of the same output of which

the largest is supercharged to 1.5 times the original output while the one here discussed has its output raised 2.7 times compared with atmospheric induction. The new engine is smaller, its specific weight is 33 pounds per horsepower compared with 61.8 pounds of the other without any increase of piston speed. The fuel consumption dropped from .380 to .313 lbs/BHP-hr (170 vs 140 g/PSeh) and lubricating oil consumption from .0045 to .0056 lbs/BHP-hr (2 to 2.5 g/PSeh) to .0011-.0018 lbs/BHP-hr (0.5 to 0.8 g/PSeh). Our daylong tests have given us a total lubricating oil consumption of 1.875 pounds per hour (0.85 kg/h) which corresponds to a specific consumption of .0014 lbs/BHP -hr when referred to output for optimum fuel consumption—BMEP = 227 psi (16 at.). Thus preliminary calculations evidently are here confirmed also. This low consumption of lubricating oil is the more important because the latter is several times more expensive than the fuel oil.

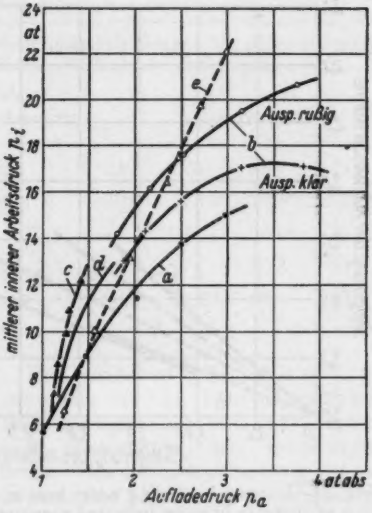
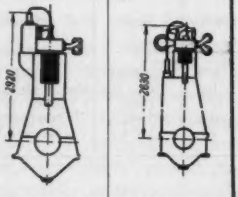


Fig. 25—Mean indicated pressures of same engines as in Fig. 24 as a function of charging pressure.

2920 = 115; 2630 = 103.5

TABLE 3—COMPARISON DATA FOR TWO ENGINES OF THE SAME OUTPUT TURBOCHARGED TO DIFFERENT CHARGING PRESSURES.



Type	G6V 40/60	K6V 30/45
Cylinder bore ..... ins	15.75	11.8
Stroke ..... ins	23.6	17.7
Weight without blower, about ..... lbs	76,000	40,000
Weight per bhp ..... lbs/bhp	61.8	33.1
Center to center distance/bore ratio	1.7	1.67
BHP per cylinder ..... bhp	195	200
BMEP ..... psi	120.7	213.0
Supercharging ..... %	50	170
rpm ..... rpm	275	375
Mean piston speed ..... ft/min	1083	1103
Fuel consumption ..... lbs/bhp-hr	.380	.313
Lub. oil consumption ..... lbs/bhp-hr	.0045-.0056	.0011-.0018
Crosshead	without	with

OPERATING EXPERIENCES

The engines test results are attractive but the operating engineer will ask—has the experience demonstrated it capable of reliable performance? How long has it been operated? Up to the time of the tests reported here the engine had run over 1000 operating hours. Many parts of its three-cylinder predecessor had been utilized in constructing the final test engine, such as the frame, the bedplate, its running parts, cylinder heads, fuel pumps, etc., only the crankshaft had of course been replaced and new cylinder liners were provided to assure an irreproachable base for wear measurement. The taken over parts had been in operation 773 hours as parts of three-cylinder engine. Thus although 1800 operating hours do not answer all questions they permit important deductions nevertheless. The post-trial inspection of all parts has shown that they were in good condition and could be reassembled without hesitation.

Another important indication of whether the increased output is within the operating conditions tolerated by the engine is the *cylinder liner wear*. Fig. 26 shows the normal growth of wear in the zone of the top ring with time—first quite fast and later slower but continuously. Its absolute value even for originally unplated rings was not unusual, but for continuous operation an improvement was desired. In other cylinders where chrome-plated rings Numbers 1 and 2 were installed originally a quite satisfactory wear of $\frac{2.36}{1000}$ inch $\left(\frac{6}{100}\text{ mm}\right)$ per 1000 working hours was obtained. Besides, the search for the optimum combination of materials can only now begin.

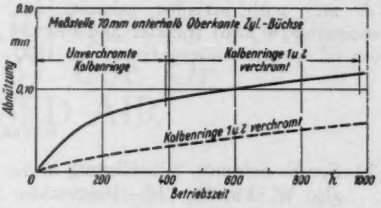


Fig. 26—Wear of cylinder liner of the high pressure charged test engine.

The engine runs, in general, softly, which, as far as the rise of combustion pressure is connected with it, was already demonstrated by the shape of indicator cards and by the numerical value of the pressure rise per degree of crankangle. The exhaust gases were, as already stated, completely invisible.

CONCLUSION

First the possible means for raising engine output in general and the nature of supercharging in particular are discussed then the constructional details are described of a new engine in which minimum fuel consumption and high brake mean effective pressure were achieved by means of an exhaust driven turbocharger by the MAN company, a recent development. The results of thorough tests are analyzed and show that an optimum fuel consumption of less than .313 lbs/BHP-hr (140g/PSeh) corresponding to a thermal efficiency of about 45% is obtained at an output of about 227 psi BMEP. This achievement is the result of a systematic and rational utilization of many thermodynamic influences, the creation of an efficient exhaust gas driven turbocharger as well as of optimum mechanical conditions. The brake mean effective pressure achieved is three times that of a

conventional atmospheric induction engine and could be raised above 284 psi (20 at.) with invisible exhaust. Investigation of heat transfer and discussion of operational experience for 1800 hours complete the picture. It may be symbolic that this new development took place in a shop designated as the actual birthplace of the Diesel engine by a suitable tablet on its wall.

References

(1) See E. Schmidt, "Einführung in die technische Thermodynamik." Berlin 1945, also W. Pflaum, "IS—Diagramme für Verbrennungsgase und ihre Anwendung auf die Verbrennungsmaschine." Berlin 1932.

(2) Soerensen, E. and Zinner, K., "Investigations to obtain a maximum thermal efficiency with Diesel engines." Premier Congres International des Moteurs a combustion interne. Vol. I, pp. 145-161, Paris, May 1951.

(3) The management was anxious to meet all requests having that in mind. Particular thanks are due to Dr.-Ing. K. Zinner, director of research for MAN Augsburg works, and to many research engineers and engine operating personnel for the tireless efforts to help during the long-lasting runs, as well as our college assistants Obering. Dipl-Ing G. Kranold and Dipl.-Ing T. Suciv for participation and assistance in the evaluation of the extensive test data.

(4) Compare with operating data of conventional supercharged engines. W. Pflaum, "Cooperation of engine and blower in supercharged diesel engines." 74th Annual Meeting of VDI at Darmstadt in 1936, also VDI special issue, Diesel Engines, VII, Berlin 1938, also K. Zinner, "Supercharging of Diesel Engines," Motortechn. Z. (1950) pp. 57-67.

(5) Eichelberg, A., "Investigations on Internal Combustion Engine Problems." Engineering 1939.

(6) Eichelberg, G., "Temperaturverlauf und Warmespannungen in Verbrennungsmotoren." VDI Forschungsheft 263, Berlin 1923.

(7) Jakob, M., und Linke, W., "Der Warmeübergung von einer wargerechten Platte an Siedendes Wasser." Forschung 1933, pp. 75-81.

(8) Pflaum, W., "Möglichkeiten und Grenzen der Drehzahl und Arbeits-drucksteigerung." Motor und Kraftstoff, Report of the 1938 Fall Meeting of VDI in Augsburg.

TESTING COMPARTMENTS FOR TIGHTNESS BY USE OF COMPRESSED AIR

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EDITOR'S NOTE

The complete article by Admiral McShane is too long for publication in a single issue of the JOURNAL. *Parts I, II, and III were published in the May 1952 issue of the* JOURNAL. *Parts IV, V, and VI are published below. Parts VII and VIII and the Appendices will appear later.*

PART TWO

REFERENCES

(a) Bureau of C & R pamphlet, "Air Tests of Compartments of Vessels of the U. S. Navy," reprint dated January, 1922.
(b) Bureau of C & R letter S-29-8(TB) EN7/A2-11 of September 20, 1933, to Commandants, all Navy Yards and all Superintending Constructors.
(c) Bureau of C & R letter A10/S29-8 (C & DC) of August 20, 1935, to Commandant, Navy Yard, Norfolk, Va.
(d) Norfolk Test Serial No. 188—"Air Tests of Compartments—Investigation and Experiments."
(e) Bureau of C & R letter S29-8 (C & DC) of June 1, 1938, to Commandant Navy Yard, Philadelphia, Pa.
(f) Bureau of Ships letter S29-8 (Dz) EN28/A2-11 of March 11, 1941, to Commandants, all Navy Yards and all Supervisors of Shipbuilding.

APPENDICES

I. Mathematical Analysis.
II. Probable Accuracy of Past Determinations of Aperture Area.

IV. LEAKAGE PHENOMENA

In the development of the equations which depict the leakage of air through apertures ideal conditions are assumed and in the formulation of equations the following practical factors were neglected:

(a) the contraction of the air in the orifices.

(b) the frictional characteristics of the orifices.

Thus, for practical employment of these relations it becomes necessary to introduce the "corrective" factor c.

From the "tightness" equation

$$A = \frac{1}{c} \frac{Vd}{nDt}$$

it is obvious that the aperture area cannot be determined until the value of c is known. However, c must be determined experimentally; a process which requires that A be known. For the apertures involved in air testing it obviously is impracticable to measure A and an impasse exists. The only way out is to ascertain values of c by inference.

In ref. (a) this problem was approached by determining the values of c for small holes of circular cross-section which, presumably, also were of straight, smooth bore and equal lengths. The results of these tests confirmed the findings of other investigations that the value of c varies with both the test pressure and the aperture area. In view of the fact that the aim in ref. (a) was to provide a single curve of pressure versus time applicable to all aperture areas the fact that c varied with area had to be compensated. Consequently, ref. (a) adopted the approximation that c would vary only as test pressure and employed the mean value of c for the circular holes tested. The "areas" determined from the subsequent tests of fittings therefore are based on the implicit assumptions that:

(a) the value of c for all apertures varies only with the test pressure.

(b) all apertures in compartment testing have values of c which vary in the same fashion as a theoretical "mean" orifice of straight, smooth bore, circular cross-section and a given length.

Neither of these assumptions is in accord with observed data and there is evidence to support the belief that errors of significant magnitude are introduced.

The pressure versus time curve of ref. (d) is based on average results for a large number of fittings. Again the hidden assumptions are the same in essence as those stated above and inherent errors are the result.

It is unfortunate that the experimental data obtained for ref. (a) are not available for analysis. There is, however, a quantity of experimental data existing in ref. (d) from which certain conclusions may be drawn. Many tests were made to determine the leakage characteristics of various types of fittings. A sample of the data in the form in which it was recorded is given below.

Pressed Panel W.T. Door 26″ × 48″	Pressure in lbs.	1	2	3	5	8	10	12	13	14	15
	Time in minutes.	30	30	30	30	30	30	30	30	30	30
	Loss in ounces.	0	0	0	0	0	0	1	1½	2½	4½
	Gain in ounces.	0	½	½	½	½	0	0	0	0	0
	Temperature °F	77	78	80	81	82	83	85	86	87	87

From these data an analysis was made in the following manner:

(a) using the pressure and observed change in pressure in the given time, the value of ntcA/V was calculated from equation (32). It will be noted in the example above that in certain tests a *"gain* in ounces" is reported. This occurred in many cases and indicates that the value of n was negative. Hence, the value of ntcA/V is negative.

(b) knowing t the value of ncA/V was determined. Inasmuch as V was the same in all cases (496 cu. ft.) the values of ncA/V are satisfactory for making comparisons of ncA. The values of ncA/V thus determined are recorded in the following tables. The values given are 10^7 times the actual values.

VALUES OF ncA/V × 10^7

Type, Size, etc.	Test Pressure, p.s.i. gauge													Notes
	¼	½	1	2	3	4	5	8	10	12	13	14	15	
CIRCULAR HOLES														
1/64" diameter	−4.8	−5.0	−2.5	−1.8	−0.8	−0.5	−0.7		3.0				2.5	
1/32" diameter	7.7	14.0	13.0	18.0	16.0	17.7	18.7		17.7				18.7	
1/16" diameter	32.7	45.6	42.4	49.7	53.7	51.0	56.4		53.7				65.0	
3/32" diameter	53.4	64.0	86.4	102	111	105	107		107				115	
1/8" diameter	241	97.0	182	221	221	230	229		232				245	
W.T. DOORS														
Pressed Panel 26" × 66"			−2.0	−1.7	0		0	0	−3.2	8.7				
Pressed Panel 26" × 66"			−4.8	−1.7	−1.3		−2.2	−1.7	−0.7	0			0	a
Pressed Panel 26" × 66"			−4.8	−1.7	−1.3		−1.0	−1.7	−4.7	−1.3			−2.7	b
Pressed Panel 26" × 66"			−4.8	−1.7	0		0	0	0	2.8	12.7	27.0		b
Pressed Panel 26" × 48"			0	−3.3	−2.7		−3.2	−2.7	0	2.8	4.3	6.7	11.3	
Pressed Panel 26" × 48"			−2.0	0	−1.3		−1.0	−0.8	−0.7	5.0	5.8	27.0		b
C.R.S. 26" × 66"			2.0	3.3	7.0	10.0	13.3							
C.R.S. 26" × 54"			0	0			1.0	7.0	16.0	35.0			13.2	
C.R.S 26" × 54"			0	0			0	0.8	4.0	11.7			9.2	
C.R.S 26" × 54"			0	0			0	1.7	3.2	5.0			9.2	

Type, Size, etc.	1	2	3	4	5	7	8	9	10	10½	11	12	15	Notes
RAISED W.T. HATCHES														
C.R.S. 30" × 30"	0	0	0	5.0	9.0									c
C.R.S. 30" × 30"	0	0	0	2.5	4.5									c
C.R.S. 30" × 30"				3.7	5.7									c
C.R.S. 30" × 30"	0	0	0		0	0		0		1.5				d
C.R.S. 30" × 30"	0	0	0		0	0		0.8		2.7				d
C.R.S. 48" × 30"	0	3.3	5.7	6.2	4.5									e
C.R.S. 48" × 30"	0	3.3	5.7	6.2	6.7									e

COMPARTMENT AIR TESTING

Type, Size, etc.	1	2	3	4	5	7	8	9	10	10½	11	12	15	Notes
RAISED MANHOLE COVERS														
Pressed Panel 15″ × 23″	-4.7	-3.3	-2.7		-2.2		0		0			0	0	f
Pressed Panel 15″ × 23″	-4.7	-3.3	-2.7		-6.7		0		0			4.3	25.0	g
Pressed Panel 15″ × 23″	0	0	0		0		0		1.5		1.5	1.3	2.7	h
Pressed Panel 15″ × 23″	0	0	0		0		0		0		0	0	0	i
Pressed Panel 15″ × 23″	-4.7	-3.3	-2.7		1.0		1.7		11.2		24.5	43.7	114	j
ESCAPE SCUTTLE														
18″ diameter, 4 dogs	0	0	1.3		3.3		3.5		4.7		4.5	2.8	6.3	
18″ diameter, 4 dogs	0	0	1.3		3.3		3.5		4.7		6.8	19.0	160	j
FLAT PLATE BOLTED MANHOLE COVER														
13″ × 18″	36.7	57.3	61.3		68.0		75.0		83.0			88.0	103	k
13″ × 18″	0	0	0		0		0		0.8			1.3	2.7	l
13″ × 18″	0	0	0		0		0		0			0	0	l

Type, Size, etc.	½	1	1½	2	3	5	8	10	12	13	15	Notes
VENTILATION CLOSING COVERS												
6″ diameter		-10.0		-3.3	0	0	3.5	4.7		5.7		
6″ diameter		-4.7		-3.3	2.7	6.7	7.0	9.5		8.3		m
8″ diameter		-4.7		0	0	0	-2.7	0	0		0	m
8″ diameter		-10.0		-3.3	-2.7	-2.2	0	0	0		0	
8″ diameter		-10.0		-3.3	-1.3	-4.3	-3.5	-3.2	-2.8		0	n
10″ diameter		0		0	0	0	0	0				
20″ diameter	-14.0	-4.7	-4.0	-7.0								
21″ × 15″	0	0	16.3	21.3								
21″ × 15″	0	0	-4.0	-3.3								
12″ × 14″		10.0		14.0	17.2	29.3	153					o
12″ × 14″		-4.7		0	-1.3	0	0	1.5		36.7		p
24″ × 12½″		-20.3		-3.3	-2.7	-2.2	0	0	0		5.3	
STUFFING BOXES												
Operating Rod, 1″		-10.0		-7.0	-5.7	-2.2	-1.7	-3.2	-2.8		-2.7	
Operating Rod, 1½″		0		-3.3	-2.7	-2.2	-1.7	-1.5	-1.3		-2.0	
Operating Rod, 1½″		-4.7		-3.3	-2.7	0	0	0	0		0	
Shaft, 2″		0		0	0	0	0	-2.3	-1.3			
Pipe, 2½″		-2.3		-3.3	-5.7	-3.3	0	-0.8	-1.3		0	
Pipe, 2½″		-4.7		-1.7	0	0	0	1.5	0.7		1.2	n
Voice Tube, 4″		-2.3		-3.3	-2.7	-3.3	-3.5	-3.2	-2.8		-2.7	
Voice Tube, 4″		0		0	0	2.2	3.5	4.0	3.7		5.3	n

Type, Size, etc.	1	2	3	5	8	10	12	15	Notes
BULKHEAD FLANGES									
2″ bolted, W.T. gasket	−4.7	−3.3	−2.7	−2.2	−0.8	0	0	0	
2″ bolted, O.T. gasket	−4.7	−3.3	−1.3	−1.0	0	0	0	0	
6″ welded, W.T. gasket	−4.7	−7.0	−4.3	−3.3	−2.7	−2.3	−2.8	−2.0	
6″ riveted, W.T. gasket	−2.3	−3.3	0	0	0	0	−1.3	−2.0	
6″ bolted, W.T. gasket	0	0	1.3	2.2	2.7	3.2	4.3	5.3	
6″ bolted, W.T. gasket	−2.3	−3.3	−4.3	−4.5	−3.5	−2.3	−2.0	−1.3	
6″ bolted, W.T. gasket	0	0	0	0	0	0	0	0	
6″ bolted, W.T. gasket	0	0	4.3	4.5	3.5	7.8	3.7	7.8	
6″ welded, O.T. gasket	0	0	0	0	0	0	0	0	
6″ riveted, O.T. gasket	0	0	0	0	0	0	−0.7	−0.7	
6″ bolted, O.T. gasket	0	0	0	0	0	0	−1.3	0	
SOUNDING TUBE AND AIR ESCAPE DECK PLATES									
	0	0	1.3	4.5	4.3	6.3	5.8	7.8	q
	0	0	0	0	0	0	0	0	r
	0	1.7	2.7	4.5	7.0	6.3	6.5	7.2	n

ELECTRIC CABLE

Type	diam. ins.	Test Pressure, p.s.i. gauge															
		Test 1*				Test 2*				Test 3*				Test 4*			
		2	5	10	15	2	5	10	15	2	5	10	15	2	5	10	15
TFPP 400	2.60	3.3	20.2	30.3	39.3	7.0	20.2	31.8	39.3	7.0	18.0	31.8	39.3	7.0	18.0	28.8	34.0
250	2.21	3.3	6.7	16.0	15.7	3.3	4.5	11.2	11.7	3.3	4.5	7.7	10.5	1.7	4.5	12.7	18.3
100	1.61	1.7	6.7	12.7	15.7	3.3	6.7	12.7	15.7	0	0	0.8	2.7	0	0	1.5	3.3
40	1.16	1.7	4.5	4.7	5.3	0	1.0	2.3	4.0	1.7	4.5	4.7	7.8	1.7	2.0	3.2	6.3
9	0.80	1.7	2.0	2.8	4.0	0	1.0	1.3	2.7	0	1.0	0.7	2.7	0	1.0	0.7	1.3
														Test 3—Repeat			
TFPA 250	2.28	24.2	39.1	51.1	57.7					25.3	38.2	49.2	61.7				
100	1.68	7.1	13.4	17.0	22.6					4.7	15.0	21.2	28.9				
40	1.23	—	0.7	2.1	5.2					—	2.6	3.1	7.0	—	0.7	2.1	2.7
9	0.88	—	—	—	0.8					—	—	1.1	2.7				
MFPA 26	1.42	1.1	3.0	9.6	11.3					2.2	4.4	9.6	14.0	2.2	6.8	9.6	15.8
10	1.04	—	0.7	1.0	1.8					—	0.7	1.0	1.8				
DLPA 400	2.51	4.7	15.0	18.1	24.4					2.2	4.4	8.4	10.4				
250	2.10	1.1	3.0	5.2	7.0					1.1	3.0	5.2	7.0				
100	1.56	—	1.3	1.6	2.7					—	1.3	1.6	2.7				
40	1.22	—	—	0.4	1.8					—	—	0.4	1.8				
9	0.95	—	—	—	0.4					—	—	—	0.4				

*Test 1—Cable passed through stuffing tube and terminating in W.T. terminal box within tank. Cable painted with 2 coats aluminum paint.

Test 2—Same as 1 with addition of 2 coats of air-drying varnish.

Test 3—Cable passed into tank through one stuffing tube and out through another with 10 ft. loop in tank. Two coats aluminum paint.

Test 4—Same as 3 with addition of 2 coats of air-drying varnish.

NOTES

a. Adjusted to leak 5 gals at 15 p.s.i.
b. Adjusted to leak 10 gals. at 15 p.s.i.
c. 8 drop bolts
d. 6 dogs
e. 10 dogs
f. 6 drop bolts
g. Adjusted to leak at 11 p.s.i.
h. 4 drop bolts
i. Adjusted to leak 3 gals. at 11 p.s.i.
j. Adjusted to leak 6 gals. at 11 p.s.i.
k. Ordinary bolts
l. Machined bolts
m. Pressure to open
n. Adjusted to leak at 15 p.s.i.
o. Strong-back not according to plan
p. Strong-back modified
q. Screws tapped through plate
r. Screws tapped through plate grommets and washers.

The only certain conclusions which can be drawn from these tests is that, in view of the large number of negative values, heat transfer was a frequent disturbing factor and its magnitude was sufficient to vitiate completely the accuracy of the determination of aperture. It is perfectly obvious from the tests on the 1/64 in. circular hole, where leakage was a certainty, that the usefulness of the remainder of the data is impaired. It is unfortunate that temperatures were not recorded at the start and finish of each test in which case the data could have been adjusted approximately to a common basis but the fact that only a single temperature was recorded and the time of its observation is unknown, render those data completely useless. In addition, the drops were recorded to the nearest quarter ounce and thus errors of large value are possible particularly in case of the low drops. Therefore, the aperture data are very, very approximate. However, they are all that are available at present and some tentative conclusions can be drawn from them.

Circular Holes—In these cases the value of A (and V) is known and, thus, the value of nc can be determined. The results are as follows:

Test Pressure	¼	½	1	2	3	4	5	10	15
1/64″ diameter	−1.3	−1.3	−0.7	−0.5	−0.2	−0.1	−0.2	0.8	0.7
1/32″ diameter	0.5	0.9	0.8	1.2	1.0	1.1	1.2	1.1	1.2
1/16″ diameter	0.5	0.7	0.7	0.8	0.9	0.8	0.9	0.9	1.1
3/32″ diameter	0.4	0.5	0.6	0.7	0.8	0.8	0.8	0.8	0.8
1/8″ diameter	1.0	0.4	0.7	0.9	0.9	0.9	0.9	0.9	1.0

Clearly, the value of c can be neither negative nor greater than unity. Therefore, values of n were variable. Consequently, the data above cannot be used for quantitative conclusions. The best that can be said is that the values of c do not vary greatly according either to area or pressure. It may also be concluded that *possibly* the same kind of aberrations, although probably not the same degree, occurred in the tests reported in ref. (a).

Apertures in fittings—It is difficult to detect any orderliness in the data for ship fittings. It does seem possible, however, that the following are indicated:

(a) gasketed diaphragms such as doors, hatch covers, manhole covers, scuttles, ventilated closing covers, etc. reach a certain critical pressure above which the value of cA will increase very rapidly due to distortion. Presumably, these fittings were tested with the pressure tending to "open" them. Had tests been made with pressure on the other side, it is possible that different leakage characteristics would have been observed.

(b) repeat tests on such diaphragms indicate the variability of such fittings under presumably similar conditions. In other words, it seems that conditions beyond practicable control would result in sizable differences in leakage characteristics.

(c) in general, where the fittings were not adjusted arbitrarily to leak, only W.T. doors, hatches and electric cables show appreciable apertures. Indeed, it looks as though these fittings alone may have been responsible in the past for the greater amount of leakage. This fact may have been solely responsible for the great number of drops much lower than "allowable" which were noticed in the past in that many of the fittings for which apertures had been "allowed" actually were not leaking.

The fact of greatest importance is that the *rate* of increase of cA with pressure is very much more rapid for all types of fittings than for circular holes. To illustrate this fact the following tabulation is offered. In the tables the data have been selected and presented in the following manner:

(a) the tests selected were those in which no negative values appear in order to choose data where temperature variations did not completely obscure the leakage.

(b) in order to show most effectively the variation of ncA with pressure the value of ncA at 5 p.s.i. has, so far as possible, been assigned as 100% and values at other pressures have been referred to that figure as a base.

There is far too great potential error in the data to make any flat statements concerning its significance. It is possible that the effect of heat transfer renders the figures completely useless. However, if the general nature of the values is about correct it can be concluded that:

(a) the variation of cA with pressure differs greatly between fittings and circular holes.

(b) the characteristics of the curve of cA versus pressure for various fittings is greatly different.

The significance of these facts is that no single scale of values of c will apply to all fittings; therefore, no single curve of pressure versus time will apply to all. These facts make it impossible to ascertain *true* apperture areas.

TEST PRESSURE (P.S.I. GAUGE)

	1	2	3	5	10	15
Circular hole, 1/32″ diameter	70	96	85	100	95	100
Circular hole, 1/16″ diameter	75	88	95	100	95	115
Circular hole, 3/32″ diameter	81	95	114	100	100	107
Circular hole, 1/8″ diameter	79	96	96	100	101	107
W.T. door, C.R.S., 26″ × 66″	15	25	53	100		
W.T. door, C.R.S., 26″ × 54″	0	0	—	100	1600	1320
W.T. door, C.R.S., 26″ × 54″	0	0	—	0	1600	3680
W.T. door, C.R.S., 26″ × 54″	0	0	—	0	1600	4600
Raised W.T. hatch, C.R.S., 30″ × 30″, 8 bolts	0	0	0	100		
Raised W.T. hatch, C.R.S., 48″ × 30″, 10 dogs	0	73	127	100		
Raised W.T. hatch, C.R.S.	0	49	85	100		
Manhole cover, pressed panel, 15″ × 23″, 4 bolts	0	0	0	0	100	180
Manhole cover, pressed panel, 15″ × 23″	0	0	0	0	0	0
Escape scuttle, 18″ diameter, 4 dogs	0	0	39	100	143	191
Flat plate manhole cover, ordinary bolts	54	84	90	100	122	151
Flat plate manhole cover, machined bolts	0	0	0	0	100	337
Flat plate manhole cover, machined bolts	0	0	0	0	0	0
Stuffing box, voice tube, 4″	0	0	0	100	182	241
Bulkhead flange, bolted, W.T. gasket	0	0	96	100	173	173
Bulkhead flange, welded, O.T. gasket	0	0	0	0	0	0
Deck plate, screws tapped through plate	0	0	29	100	140	173
Deck plate, grommets and washers	0	0	0	0	0	0
Deck plate, adjusted to leak at 15	0	38	60	100	140	160
Electric cable, TFPP 400, 2.60″ diameter	—	16	—	100	150	194
Electric cable, TFPP 400, 2.60″ diameter	—	35	—	100	157	194
Electric cable, TFPP 400, 2.60″ diameter	—	39	—	100	177	218
Electric cable, TFPP 400, 2.60″ diameter	—	39	—	100	160	189
Electric cable, TFPP 250, 2.21″ diameter	—	49	—	100	239	234
Electric cable, TFPP 250, 2.21″ diameter	—	73	—	100	249	260
Electric cable, TFPP 250, 2.21″ diameter	—	73	—	100	171	233
Electric cable, TFPP 250, 2.21″ diameter	—	38	—	100	282	406
Electric cable, TFPP 100, 1.61″ diameter	—	25	—	100	189	234
Electric cable, TFPP 100, 1.61″ diameter	—	49	—	100	189	234
Electric cable, TFPA 250, 2.28″ diameter	—	62	—	100	131	148
Electric cable, TFPA 250, 2.28″ diameter	—	66	—	100	129	162
Electric cable, TFPA 100, 1.68″ diameter	—	53	—	100	127	169
Electric cable, TFPA 100, 1.68″ diameter	—	31	—	100	141	193
Electric cable, MFPA 26, 1.42″ diameter	—	37	—	100	320	378
Electric cable, MFPA 26, 1.42″ diameter	—	50	—	100	218	318
Electric cable, DLPA 400, 2.51″ diameter	—	31	—	100	121	163
Electric cable, DLPA 400, 2.51″ diameter	—	50	—	100	191	236

The very rapid increase of ncA values given above probably indicates that, at least for the types of fittings tabulated, the value of A increases as the pressure increases. Probably in the case of many fittings distortion occurs at higher pressures to increase the value of A. Very likely in the case of rivets, riveted joints and welds the values of A would remain constant but for many other sources of leakage this condition will not exist.

Thus, we are confronted by an apparent inability to ascertain true aperture values and thereby have no fixed way of evaluating tightness with certainty. The only immediately apparent recourse is to rely on values of cA as a measure of tightness. This value will be termed the "fictive" aperture and is proposed as the criterion of tightness.

The *rate* of air leakage depends on the value of cA and its adoption as the cri-

terion of compartment tightness implies that it also is an index of the *rate* of leakage of liquids; i.e., that c for liquids will vary in about the same fashion as for air. This assumption is not accurate but probably it will introduce errors of less magnitude than the assumptions previously made in the endeavor to get true values of A.

The elimination of quantitative values of c introduces an important consideration which should not be overlooked. The attempt in ref. (a), and all other methods which followed its lead, was to obtain an aperture value which presumably remained fixed regardless of pressure. Theoretically the "allowable" apertures of that method could be used at any test pressure. The indications of the data given above are that such a scheme is possible only if several sets of c values were employed for fittings of various classes. For instance, if careful experiments on a number of fittings indicated that they could be classed in several groups according to the variation of cA with pressure it would be possible to assign several scales of multiplying factors to show the proper relative values of cA at various pressures. Then, instead of applying a single value of c at a given pressure to all fittings the appropriate value would be used depending on the type of fitting. The result for a given compartment would be a "fictive" aperture value applicable to the given test pressure. Such refinement seems unnecessary and, as will be pointed out later, there are means of criticizing compartment tightness which eliminate the adoption of such a burdensome process.

For the remainder of this discussion the factor c will, as a general rule, be omitted from further consideration and, unless otherwise indicated, where A is used it will mean "fictive" aperture.

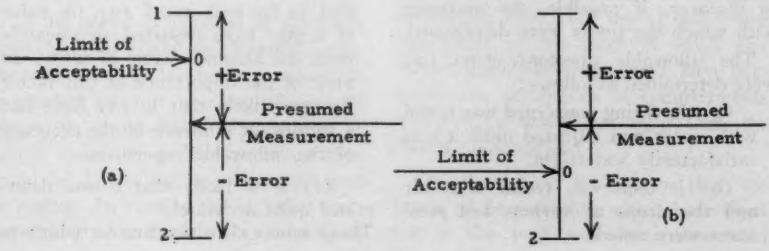
V. ACCURACY IN AIR TESTING

Tightness air tests have for their primary purpose the rectification of apertures which have been discovered by sound, feel, soap bubbles or other methods. It would be possible to rely solely on this method but, in view of the importance of tightness, it has been considered necessary to supplement this primary method by *measuring* the degree of tightness attained. Inevitably, then, there arise two problems:

(a) the assignment of acceptable limits of tightness

(b) the measuring of the tightness of specific compartments and deciding whether they meet the requirements or whether further corrective work should be performed to improve them.

The question of accuracy must be considered in both problems but it is of particular importance in the second. If the method by which tightness is measured is highly inaccurate one of two very undesirable conditions must arise in many instances. These conditions are illustrated by the diagrams.



If the inaccurately measured value is as shown in (a) the compartment will be accepted whereas the actual value may have been between 0 and 1, and the compartment is less tight than desired. If the condition is as shown in (b) corrective work will be demanded, frequently involving considerable expenditure of time and money, whereas the actual value may have been between 0 and 2, and the compartment complies with the requirements. Obviously, the question of accuracy is important.

It is not quite right to dismiss the question on the grounds that experience has shown results in the past to be good enough until it is determined how good they were. It is entirely possible that they may have been "too good" and that inordinately rigorous requirements prevailed. It is pertinent to examine what past requirements have been.

Accuracy of past limits of acceptability—Until March, 1941, the assignment of acceptable limits of tightness followed the method of ref. (a) and consisted of

(a) determining the "allowable" apertures for the structure and fittings in the compartment concerned.

(b) calculating the volume of the compartment.

(c) assigning the time period to be used.

(d) calculating, on the basis of the above factors, the "allowable" drop. This step was performed either by the use of the "pressure vs. time" curve of ref. (a) or by the "K" curve of ref. (b).

Each of these steps must be examined to discover, if possible, the accuracy with which the limits were determined.

The "allowable" apertures of ref. (a) were determined as follows:

(a) the fitting concerned was tested with water and adjusted until it was satisfactorily watertight.

(b) it then was tested with air and the drops at various test pressures were noted.

(c) its "aperture" then was calculated from the pressure versus time curve of ref. (a). This process, in effect, is the solution of the equation.

$$A = \frac{Vd}{Dcnt}$$

In Appendix II the above equation is analyzed from the point of view of the accuracy which has been attained in the past practice of compartment testing. By a similar analysis, which is summarized below, a rough measure of the probable accuracy of the "allowable" apertures in ref. (a) can be obtained.

(a) The volume of the test tank probably was accurately known.

(b) the observed drops apparently were relatively large and, presumably, a number of tests were made. Thus, the drop probably was measured with reasonable accuracy.

(c) there is no way to determine the accuracy of D but, in any event, the error between the actual and assumed conditions probably was negligible.

(d) the probable accuracy of the c values adopted in ref. (a) is discussed in Part IV. It is likely that the values of c which were adopted were incorrect. Thus the "true" areas given in ref. (a) probably are incorrect. It seems probable that the *relative* values of A given in ref. (a) are subject to great errors due to the probability that the change of c with pressure is not the same for all fittings although, in ref. (a) it was assumed to be.

(e) there is some slight evidence that, in the tests in ref. (a), the value of n may have departed significantly from the assumed value of unity. In view of the importance of this factor it seems likely that it may have had a significant influence on the accuracy of the "allowable" apertures.

(f) it is likely that t was determined quite accurately.

There now exists no data on which to

make a guess at the quantitative accuracy of the "allowable" apertures of ref. (a). It would seem that the errors are certainly as large as ± 10% and probably very much greater than that. Many of the values given in ref. (a) were continued in use as long as the method of ref. (a) was used.

As time passed the need arose for allowable apertures for fittings not listed in ref. (a). How these were obtained is not quite clear but it is likely that many of them were estimates based on the data in ref. (a). There is some reason to suspect that some "allowable" apertures used in later years were taken from ref. (d). This is probably true of electric cable fittings. The practice of using ref. (d) values would have led to very large errors because those values were intended for use with the "K" curve of ref. (d) which varies materially from that of ref. (b).

In summary, it would appear that the "allowable" aperture values which were used were subject to large errors and the appearance of greater accuracy engendered by their small magnitude is illusory.

The next step in the process of assigning the "allowable" drops, namely, the calculation of compartment volume, is discussed in Appendix II. It is likely that the calculated volumes always were higher than actual volumes due to neglect of "permeability."

The assignment of the time of the test could not involve any inaccuracies.

The calculation of allowable drops by the method of ref. (a) required that values be picked from a curve which did not lend itself to accuracy. In September, 1933 the bureau promulgated ref. (b) with its "K" curve as a means of determining allowable drop. This curve introduced further errors into the problem. The true "K" values compared with the values given by ref. (b) are as follows:

p	1	2	3	4	6	8
Correct value	2.38	3.64	4.46	5.20	6.44	7.52
Ref. (b)	2.10	3.10	3.85	4.50	5.65	6.61
Error %	—12	—15	—14	—13	—12	—12

These errors are on the conservative side in that they would result in a lower "allowable" drop than was considered acceptable. Thus, as far as the calculation of "allowable" drops was concerned, the accuracy would depend on the method used as well as the accuracy of the factors which entered the calculations.

In the actual assignment of allowable drops another factor was bound to enter the picture. While the drop could be calculated to any number of decimal places it would have been useless to assign values of greater precision than could be determined by the gauge used in testing. In view of the fact that mercury gauges are used in the majority of cases allowable drops seem commonly to have been assigned at the "next higher" half ounce. However, at least one yard assigned allowable drops in increments of eighths of an ounce although it is difficult to understand how a mercury gauge could have been read so closely unless some kind of auxiliary device was used. The use of half ounce increments in the cases of small calculated allowable drops would have permitted the acceptance of compartments which were much less tight than the theory contemplated.

On the whole it appears that the limits of acceptability which have been set in the past may have demanded

tightness which was considerably greater than necessary in some cases and may have permitted the acceptance of many others which were much less tight than was desired. The belief that modern construction had made the attainment of much greater tightness possible (which led to the investigation in ref. (d), and the adoption of the present criteria) may have been due, in part, to the assignment of limits of acceptability which, due to accumulated errors, were much too generous.

Accuracy of the determination of tightness—A mathematical analysis of the probable accuracy with which compartment tightness has been determined in the past and still is being determined is presented in Appendix II. The factors upon which accuracy depends are so many and are so variable that it is impossible to assign a definite probable error to any given set of recorded data. The best that can be done is to calculate the possible extremes of error which are quite likely to be encountered in practice. On this basis the following quite tenable conclusions may be drawn. Positive errors indicate *calculated* apertures greater than actual; negative errors mean *calculated* apertures smaller than actual.

(a) errors due to assumed standard values for physical constants, atmospheric pressure and due to inaccurate time measurements are of the order of $\pm 9\%$.

(b) errors due to the calculation of compartment volume may be expected to be about -5%.

(c) errors due to the inaccurate determination of d and to the assumption that n is unity may be expected to be:

(1) least at high pressures and large drop values. The errors probably range between 0 and $+20\%$.

(2) greatest at low pressures and small drop values when they may range from -100% to $+180\%$. An error of -100% means a completely unreliable test with a *recorded* drop of zero and a compartment which is thus indicated to be bottle-tight; its actual tightness is completely unknown. Thus a recorded drop of "zero" does not necessarily represent bottle-tightness and, indeed, it may seldom indicate that condition.

On the whole it would seem that apertures calculated from past test data would frequently be from $\pm 20\%$ to $\pm 50\%$ in error and at times, particularly at low test pressures, would be $\pm 100\%$ in error. "Plus" errors would seem, in general to be of greater magnitude than "minus" errors. This situation certainly promotes safety but it also involves much useless work in the correction of apertures which are presumed but do not actually exist.

Appendix II reveals also that the major sources of error have been:

(a) the indeterminate effect of heat transfer

(b) insufficient accuracy in the determination of drop.

By analogy, we have been using a rubber foot rule graduated in sixteenths of an inch to measure the dimensions of holes in which variations of hundredths of an inch were significant. We have not known how much the rule was expanded or compressed and, after we applied it to the hole, we have had to guess at the dimensions.

In addition to the factors susceptible of mathematical analysis there are certain practices followed by test personnel which may have further jeopardized accuracy. One tester who was interrogated stated that he considered "small" drops of about ¼ oz. to be negligible and therefore recorded them as "zero." Several expressed the belief that "there is no such thing as a bottle-tight compartment" and one man carried this view into practice by recording all "zero" drops as ⅛ oz. These may be isolated

examples but in view of the variety of erroneous opinions concerning the significance of drops, it is possible that they represent fairly common practices.

When demands for great tightness are issued to test personnel there is a temptation to adopt methods which, although perhaps only slightly unethical, are decidedly prejudicial to accuracy. For instance, it is interesting to note the effect of small angles of tilt on a mercury gauge. Suppose the gauge hangs truly vertical at the start of a test but at the conclusion of the test is tilted only 5°. The true pressure at the end is p ozs. but the gauge reads h ozs.

$$h = p \sec 5° = 1.004\ p$$

The tilt results in a "steal" of 0.004 p ozs. The result at various test pressures is shown in the table. It isn't difficult to "steal" a quarter or a half ounce, particularly at the higher pressures. There is reason to believe that some test gangs are cognizant of this practical application of trigonometry. Undoubtedly there are other esoteric practices employed in certain cases and a thorough investigation of the problem might reveal some interesting facts bearing on the accuracy of recorded data.

p		0.004p
lbs.	*ozs.*	*ozs.*
1	16	.06
3	48	.19
5	80	.32
7	112	.45
9	144	.58

These large probable errors in the past determinations of compartment tightness point to two pertinent conclusions:

(a) the recorded data for past tests probably contains errors of large magnitude in many cases and compartments which logically could have been expected to exhibit about the same degree of tightness should show wide variations in recorded results. Thus, comparisons between similar compartments or sister ships or building yards must be made liberally.

(b) many compartments probably have been accepted in the past which were measurably less tight than has been considered acceptable; in many others corrective work possibly was undertaken which was unnecessary. Neither is a desirable condition.

In the case of compartments which, through errors in the observed drop, were passed with total aperture area greater than was considered desirable, it may be argued that this does not constitute a serious situation. If the compartments in question were searched carefully and all discernible leaks were corrected, it may be said that the remaining apertures were of such small individual area that they were negligible. If this argument be employed it naturally follows that the observation of pressure drop is a waste of time and should be abolished. On the contrary, it would seem that, in consideration of the great importance of adequate tightness, the success of the primary search for leaks should be established by observing the drop in pressure. However, if compartments are to be accepted or rejected on the basis of such observations, it is unquestionably desirable that the drops be measured with the best practicable accuracy.

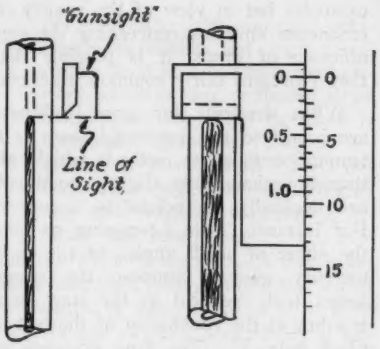
Greater accuracy is particularly desirable if it will reduce the amount of useless corrective work. The cost, in money and time, of testing compartments is large. If a reasonable proportion of this work is expended to attain an unessential degree of tightness the result is a large waste in time and labor. Greater accuracy would promote the passing of compartments when they were "tight enough" and would reduce the efforts to make them "as tight as possible." In view of the fact that many in the past probably have

been passed which were "not quite tight enough" the criterion of bottle-tightness does not appear to be generally necessary.

Methods of increasing accuracy—If laboratory accuracy were the standard to be sought the air test of a compartment would involve the use of a battery of barometers, wet and dry bulb, thermometers and other instruments. But laboratory precision is not necessary. As stated previously the chief, and only significant offenders against accuracy are inadequate precision in the determination of drop and the indeterminate effect of heat transfer. The reduction of errors due to these conditions is the most promising and effective approach to satisfactory accuracy.

The absolute probable error in drop determination is a constant value independent of the total value of the drop. In other words, it is just as likely that an *error* of ½ oz. will exist in a recorded value of drop of zero as when the recorded value is 2 ozs. Therefore the *proportionate* error may be reduced by increasing the value of the drop. From this point of view it is desirable in the test of a given compartment to increase the test pressure and the time of observation to the greatest practicable values. On the other hand, an increase in these factors promotes increased aberrations due to heat transfer unless steps are taken to eliminate this source of error.

Errors in drop readings with the present mercury gauges arise because of poor or inaccurate scale markings, the difficulty of determining the exact height of the sharply curved meniscus of the mercury column when read without auxiliary aid and the distance between the mercury column and the scale which gives rise to parallax errors. All of these sources of error could be reduced by the use of a simple auxiliary sighting device as sketched here. The sighting

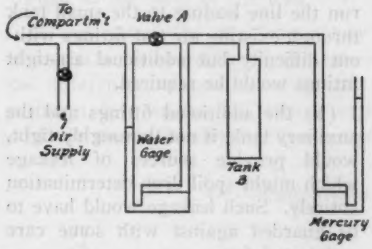


bar is provided with a "gunsight" to eliminate parallax errors. The scale is graduated in ounces and the vernier in tenths of an ounce. The device should be fitted with a screw to permit close control of the vernier movement and should be arranged to clamp to the gauge at any point. The test pressure (which is not required to be determined with great precision) should be read on the ordinary scale attached to the gauge itself. Prior to the start of the test the auxiliary device should be attached to the gauge with the sight bar somewhat above the meniscus and the bar lowered until the line of sight is tangent to the top of the meniscus, at which point the initial reading is taken. A similar procedure at the end of the test period gives the final reading and the difference is the drop in tenths of an ounce. The use of such a device might result in fewer "zero" drops being recorded.

A variety of other sensitive pressure recording devices probably is available but the manometer type of gauge now employed probably is as simple, cheap, rugged and reliable as any used and has the great advantage that test personnel already are familiar with it. The sensitivity of such a gauge is improved

by the use of a fluid lighter than mercury. For most test pressures a very light fluid is not practicable for the measurement of *total* test pressure because of the length of the gauge necessary, but it may be used in a "differential" gauge to measure *changes* in pressure. The desirability of such a gauge was recognized in ref. (d) and one type was tested but proved impractical.

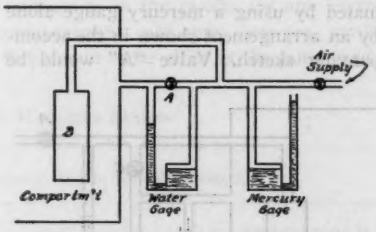
Another and simpler type of differential gauge, shown diagrammatically in the accompanying sketch, was tried by



the author. The water gauge was similar in type to the mercury gauge. While the compartment was being charged with air, valve A was kept open and the test pressure was determined by the mercury gauge in the usual manner. At the same time the tank B was being charged to the test pressure. When it was desired to determine the drop, valve A was closed thereby maintaining the *initial* pressure on the well of the water gauge. As the compartment pressure decreased due to leakage the water *rose* in the gauge glass and the height of rise determined the drop in pressure. With such a gauge the volume of the tank B must be relatively much larger than the volume of the gauge glass in order that as the water rises changes in the volume of the air in tank B and the associated tubing will result in negligible changes in tank pressure. In the gauge described here the tank volume was about ¼ cu. ft.

The advantage of such a gauge is that with distilled water as the fluid the one ounce marks are about thirteen times as far apart as on a mercury gauge. Thus drops of 0.05 oz. could be read as easily and as precisely as drops of ½ oz. on a mercury gauge.

The difficulty with such a gauge is that the tank B constitutes a very sensitive thermometer and small changes in ambient temperature measurably affect the height to which the water rises in the glass of the water gauge. However, this characteristic can be put to excellent use as shown in the diagram. By



putting tank B inside the compartment under test and closing A at the start of the test the following actions will take place:

(a) if the temperature in the compartment remains steady the initial pressure will be maintained on the well of the water gauge and the rise of water will measure the leakage drop.

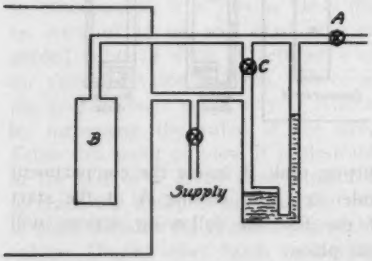
(b) if, during the test, the temperature of the air in the compartment *rises* the pressure in "B" will rise and the water will rise correspondingly higher. Note that a rise in temperature results in a value of n for the compartment air less than unity and drop alone is too small when measured by the usual means. Thus the

action of Tank "B" is to correct the drop to show what it would be if isothermal ($n = 1$) conditions had prevailed.

(c) if the compartment air temperature decreases the pressure on the well of the water gauge decreases and the drop again is shown as though the condition were isothermal.

Thus, such a device would record drops much more accurately than at present and the drops so recorded would automatically be compensated for temperature changes. They would be the drop referred to isothermal expansion, i.e., the drop due solely to leakage.

The same general conditions of temperature correction could be approximated by using a mercury gauge alone by an arrangement shown in the accompanying sketch. Valve "A" would be



open while the compartment was being charged and Valve "C" closed. When drop was to be determined, Valve "A" would be closed and Valve C cracked to charge the tank. This latter operation would drive the mercury back to zero. Then by closing Valve C the subsequent movement of the mercury column would show the leakage drop. This scheme, however, lacks the sensitivity afforded by a lighter fluid.

As proposed here, all drops would be those which would have occurred had the expansion been isothermal.

There are certain objectionable features to the use of differential gauges as proposed above. They are:

(a) the use of a tank within a compartment adds to the time of setting up for a test. It would be possible to run the line leading to the small tank through existing air test fittings without difficulty but additional air-tight fittings would be required.

(b) the additional fittings and the auxiliary tank, if not thoroughly tight, would provide sources of leakage which might spoil drop determination entirely. Such leakage would have to be guarded against with some care and tested for.

(c) a water gauge would be unsatisfactory in freezing weather and a low freezing point fluid should be used. This would require care in maintaining the proper specific gravity. The fluid should not be oily; glycerine is not satisfactory. Pure alcohol is probably the best fluid.

None of these difficulties places the method beyond the bounds of practicability but they are additional objections which must be accepted to gain greater accuracy.

VI. STANDARDS OF TIGHTNESS

It is well known that disparities exist between the test results of presumably similar compartments in sister ships but the magnitude of these variations is not generally recognized. The following tabulations are typical. The tables give an analysis of the completion test data for the ammunition and magazine spaces on sixteen sister destroyers built in eight different yards. The test pressure was 2 p. s. i. gauge in all cases. The *allowable* drop was the same for corresponding compartments on all ships.

The test data have been analyzed in three ways:

(a) in order to put all of the observed drops on a common basis they were expressed as percentages of their corresponding allowable drop. This measures the degree with which the allowed standards were met.

(b) expressed in the ratio A/V. This measures the "index" of tightness.

(c) expressed as the total aperture area which existed in the compartment at the time of the test.

In each tabulation the "consistency" of results is measured by the average of the differences, compartment by compartment, between the two ships at each yard. It should be noted that the two ships at a given yard could have a low order of tightness and still show great consistency.

From any viewpoint the greater apparent tightness attained by Yards D to H is noticeable; the performance of Yard F is remarkable. In contrast, the tightness attained by Yards A to C, although it complies in all cases with the requirements, is relatively poor.

A similar tabulation comparing seven similar CL's built at four different yards is also given. Again there is shown a decided lack of uniformity and again one yard reports remarkably good results.

In all cases there is a marked lack of uniformity in the results for any given compartment, not only between yards but also in sister ships at the same yard.

DD—Ammunition and Magazine Spaces

Comp't.	Vol.	Allowable Drop	Ratio of Observed Drop to Allowable Drop, %															
			Yard A		Yard B		Yard C		Yard D		Yard E		Yard F		Yard G		Yard H	
A–405	504	3½	0	7	14	0	57	0	0	29	0	0	0	0	0	0	0	0
406	557	3	100	67	17	100	50	0	0	17	17	17	0	50	0	0	17	33
407	1447	3½	29	14	43	0	43	29	21	14	21	0	0	0	0	0	0	0
408	685	8½	29	6	6	35	0	12	0	6	0	12	0	0	6	0	12	24
D–301	1136	7½	13	27	13	3	20	23	0	0	0	7	0	0	13	7	0	13
302	1692	3½	0	14	0	14	43	29	0	0	0	0	0	0	0	29	0	14
303	481	3	100	33	67	33	17	0	50	0	17	33	0	0	0	0	50	67
304	1034	2½	0	80	0	20	60	40	0	0	0	0	0	0	0	40	10	30
305	453	6½	0	46	0	8	54	15	0	0	0	0	0	0	15	0	0	4
307	458	3	0	0	17	8	17	17	0	0	0	0	0	0	33	0	0	0
308	409	3	33	0	0	0	67	0	0	0	0	0	0	0	0	50	0	0
Average			28	27	16	20	39	15	6	6	5	6	0	5	6	11	8	17
Average Consistency			30		24		27		10		5		5		16		9	

DD—Ammunition and Magazine Spaces

Comp't.	Vol.	Value of A/V × 10^7																
		Allowable	Yard A		Yard B		Yard C		Yard D		Yard E		Yard F		Yard G		Yard H	
A–405	504	74	0	5	10	0	42	0	0	21	0	0	0	0	0	0	0	0
406	557	64	64	42	10	64	31	0	0	10	10	10	0	31	0	0	10	21
407	1447	74	21	10	31	0	31	21	15	10	15	0	0	0	0	0	0	0
408	685	176	53	10	10	64	0	21	0	10	0	21	0	0	10	0	21	42
D–301	1136	155	21	42	21	5	31	36	0	0	0	10	0	0	21	10	0	21
302	1692	74	0	10	0	10	31	21	0	0	0	0	0	0	0	21	0	10
303	481	64	64	21	42	21	10	0	31	0	10	21	0	0	0	0	31	42
304	1034	53	0	42	0	10	31	21	0	0	0	0	0	0	0	21	5	15
305	453	143	0	64	0	10	74	21	0	0	0	0	0	0	21	0	0	5
307	458	64	0	0	10	5	10	10	0	0	0	0	0	0	21	0	0	0
308	409	64	21	0	0	0	42	0	0	0	0	0	0	0	0	31	0	0
Average		91	22	22	12	17	30	14	4	5	3	6	0	3	7	8	6	14
Average Consistency			26		20		21		7		5		3		12		8	

The disparity of results in these presumed sister ships may be accounted for in part by the inherent inaccuracies discussed in Part V and, in part, by differences in the details of design and construction but the striking differences in average results seems to point towards differing *standards* of tightness at various yards. It may be that some yards aim for *air* tightness as the customary standard while others set *water*-tightness as the goal and consider a compartment acceptable when the observed drop is somewhat less than the allowable drop.

There is a great difference in the labor and cost of corrective work required by the two standards. If air tightness is the end in view the search for leaks must be more rigorous and the corrective work more complete than seems required for watertightness. It would appear, considering the example above, that Yards A to C could have attained a greater degree of air tightness in the compartments in question had they expended further effort but, presumably, the standards of tightness at those yards did not demand it.

The original intention of ref. (a) was to set watertightness as the criterion of acceptability. At a much later date, however, this appears to have been changed and the following excerpt from a bureau letter on the subject is pertinent:

> "It is not the intention (of the bureau) that the pamphlet 'Air Testing of Compartments' nor the instructions based on it be accepted as the

DD—AMMUNITION AND MAGAZINE SPACES

Comp't.	Vol.	Value of A × 10^4																
		Allowable	Yard A		Yard B		Yard C		Yard D		Yard E		Yard F		Yard G		Yard H	
A–405	504	44	0	3	6	0	25	0	0	13	0	0	0	0	0	0	0	0
406	557	36	36	23	6	36	17	0	0	6	6	6	0	17	0	0	6	12
407	1447	108	30	15	45	0	45	30	22	15	22	0	0	0	0	0	0	0
408	685	100	31	6	6	37	0	15	0	6	0	15	0	0	6	0	15	25
D–301	1136	172	24	48	24	6	35	40	0	0	0	11	0	0	24	11	0	24
302	1692	128	0	17	0	17	52	36	0	0	0	0	0	0	0	36	0	17
303	481	31	31	10	20	10	5	0	15	0	5	10	0	0	0	0	15	20
304	1034	54	0	42	0	10	31	21	0	0	0	0	0	0	0	21	5	15
305	453	65	0	29	0	5	34	10	0	0	0	0	0	0	10	0	0	2
307	458	29	0	0	5	2	5	5	0	0	0	0	0	0	10	0	0	0
308	409	26	9	0	0	0	17	0	0	0	0	0	0	0	0	13	0	0
Average		72	15	18	10	11	24	14	3	3	3	4	0	2	5	7	4	10
Average Consistency			18		16		14		4		5		2		10		7	

standard for testing, but rather that the degree of tightness so determined be the absolute minimum which under any case is acceptable.—It is considered that the cardinal principle in compartment testing is that the source of leakage be determined and, wherever this leakage is due to imperfections or poor workmanship, corrective action be taken. The practice of searching the boundaries as fully as practicable with soap and water when a compartment is under air test should be standard and, regardless of the pressure drop attained, leaks should be reduced or eliminated wherever practicable."

This is a clear statement of the fact that *air*-tightness is the standard to be employed.

The records of past tests do not indicate that all yards have been working toward this goal. Almost certainly, part of the wide differences in test results have been due to differing standards.

Regardless of the method by which *minimum acceptable* degrees of tightness are determined this question of standards still remains to be answered. The presently existing requirements as specified in ref. (f) are more rigorous in their demands than those previously employed but they do not imply that air-tightness is the end to be sought and varying standards will continue to exist. Until the policy is stated clearly it is likely that wide differences in apparent tightness will continue.

It is debatable whether the standard of tightness should be an air-tight com-

CL—Ammunition and Magazine Spaces

Comp't.	Vol.	Value of A/V × 10^7								Value of A × 10^4							
		Allow-able	Yard I		Yard J			Yd. K	Yd. L	Allow-able	Yard I		Yard J			Yd. K	Yd. L
A–207	877	13	7	0	0	0	7	0	0	11	6	0	0	0	6	0	0
419	250	38	20	6	20	20	28	20	0	10	5	2	5	5	7	5	0
503	2030	29	15	0	15	15	8	15	0	59	30	0	30	30	16	30	0
507	4080	15	8	8	15	0	8	8	0	61	33	33	61	0	33	33	0
511	816	29	8	8	4	8	8	8	0	24	7	7	3	7	7	7	0
512	816	29	8	8	0	8	22	8	0	24	7	7	0	7	18	7	0
515	2540	22	8	0	18	0	0	8	0	56	20	0	46	0	0	20	0
516	1670	22	0	8	4	15	8	8	0	37	0	13	7	25	13	13	0
517	1670	22	0	8	4	0	0	8	0	37	0	13	7	0	0	13	0
531	1650	22	15	8	4	8	15	15	0	36	25	13	7	13	25	25	0
521	682	67	15	18	22	29	8	8	10	46	10	12	15	20	5	5	7
522	682	67	15	0	18	29	8	22	0	46	10	0	12	20	5	15	0
523	740	45	8	8	8	15	8	8	0	33	6	6	6	11	6	6	0
524	740	45	15	8	8	15	—	8	0	33	11	6	6	11	—	6	0
527	2075	22	0	15	8	8	0	8	0	46	0	31	17	17	0	17	0
528	1220	29	8	8	15	15	8	8	0	35	10	10	18	18	10	10	0
605	696	22	8	8	11	0	0	8	0	15	6	6	8	0	0	6	0
605½	300	36	0	0	0	24	0	13	0	11	0	0	0	7	0	4	0
606	991	24	13	24	36	0	13	13	0	24	13	24	36	0	13	13	0
608	1248	22	15	15	11	0	0	8	0	27	19	19	14	0	0	10	0
D–302	153	49	0	0	13	13	13	24	17	8	0	0	2	2	2	4	3
502	1380	15	8	4	15	8	15	4	0	21	11	6	21	11	21	6	0
503	506	24	—	6	20	0	0	8	0	12	—	3	10	0	0	4	0
504	660	22	8	4	4	15	4	13	0	15	5	3	3	10	3	9	0
505	159	62	24	13	36	0	6	24	0	1	0	0	0	0	0	0	0
507	4960	15	8	4	4	0	8	8	0	74	40	20	20	0	40	40	0
509	426	29	15	22	41	76	76	18	5	12	6	9	17	32	32	8	2
510	426	29	15	0	—	60	76	15	2	12	6	0	—	26	32	6	1
511	2208	15	0	8	8	15	15	8	0	33	0	18	18	33	33	18	0
Average		30	9	8	13	14	12	11	1	30	10	9	14	11	12	12	0
Average Consistency			6		14*			—	—	—	7		14*			—	—

* Based on greatest difference between any two of the three values.

partment. In the case of tanks and voids it probably is reasonable but for other spaces which are expected to be flooded only in case of emergency or damage there is room for argument. Apertures which will result in lively air leakage will, in many cases, leak negligible amounts of water and should be no cause for concern. This stand should not be interpreted to mean that poor design, careless workmanship or indifferent search for leaks should be acceptable. There should be no elimination of a careful search of all boundaries of a compartment to discover and correct all the air leaks which are discernible but once this has been done and the pressure drop has been brought below a reasonable value, it is safe to assume that the remaining air leaks are so scattered and of such small individual area that they are negligible and further search to discover and rectify them, solely in the effort to attain "zero" drops, would add little, if anything, to *useful* tightness.

The proper aim would appear to be the adoption of reliable and precise means to determine the leakage drop and the ascertainment and employment of the best practicable criterion of *useful* tightness. The latter should be no more rigorous than past experience has shown to be required.

EDITOR'S NOTE: *A previous portion of this article appeared in the May, 1952 issue of the* JOURNAL. The *remainder of it will be published in a subsequent issue.*

VIBRATIONS IN MARINE ENGINEERING

ACKNOWLEDGMENT

This article surveys the history of hull and machinery vibration, and discusses various aspects of the subject. It was written by A. E. FOTHERGILL of Scotts' Shipbuilding and Engineering Co., Ltd., and the complete article was published in Volume 95, Part 4, of the "Transactions of the Institution of Engineers and Shipbuilders in Scotland." Excerpted.

INTRODUCTION

More than a century of progress in the art of ship propulsion has resulted in the subject of this paper becoming one of ever increasing importance to engineers and shipbuilders alike. New types of propelling machinery, new methods of ship construction, higher speeds and more exacting standards for safety and human comfort have all brought in their train at some time or other problems in vibration requiring the serious consideration of designers and technical experts. Much light has been thrown on the subject in recent years and there is a clearer understanding on many points although the methods available for dealing with certain important aspects are still in an unsatisfactory state which current research is directed at improving. From the designer's point of view it must be said that there is a great need for correlating the accumulated data on the subject, as the scope has become very broad. A voluminous text-book would be required to cover all the ground, and there appears to be none yet published dealing specifically with the marine side, which has many basically different features compared with other branches of engineering.

Examination of the papers included in the Transactions of this Institution since its inception in 1857 reveals the surprising fact that the subject has received but scant attention, only two or three efforts being related thereto over that long period and these dealt only with particular phases.

Many notable contributions to knowledge of the subject have, however, originated in Great Britain, and some valuable pioneer work has also been done abroad. The paper therefore deals with the subject mainly in a general and historical manner, introducing a few detailed practical examples which are considered to be of common interest.

VIBRATIONS MET IN PRACTICE

Vibrations met in marine engineering may be divided broadly into two main classes: firstly, those occurring within the main machinery and not appreciably affecting surrounding parts. These may affect safety, reliability, and efficiency; they may be caused by internal disturbances such as those arising from uneven turning effort in reciprocating engines, unsuitable proportions, or inaccurate production (for example, tooth errors in transmission gearing, unbalanced rotors, etc.), or they may be due to external influences, as for instance the varying resistance to the propeller drive, which is invariably present in some degree. Secondly, vibrations caused by the machinery with little or no loss of performance, may set up objectionable effects in the spaces or structure of the hull. These may affect the ship as a whole, or local parts only, and may be manifest as relative periodic movement or as unpleasant noise. The root causes include uneven turning effort, unsuitable balance of reciprocating engine parts, unbalanced rotating parts, variable propeller reaction, and pulsation of induction and exhaust.

Considering the types of vibration coming within the first category, there is the possibility of:

(*a*) *Torsional Vibrations* due to uneven turning effort or varying propeller resistance or both. These are most troublesome to deal with on main propelling machinery of the reciprocating type. Sometimes there is external evidence of the condition, such as the rattling of chain drives and gear drives on the engine and increased noise and clicking of teeth on geared transmissions, but severe torsional vibrations of high periodicity, which may ultimately lead to shaft failure, can occur without much external indication.

(*b*) *Longitudinal Vibrations* set up through varying propeller thrust, the combined effect of the uneven turning effort and the blades working in a non-uniform wake stream.

(*c*) *Lateral Vibrations* as evidenced by the whipping of shafts in the transmission system. There is a particular type of lateral vibration associated with the bending of the screw-shaft while rotating, on account of the uneven distribution of thrust over the propeller disc. This will be referred to later.

(*d*) *Transverse Engine Vibration.* In reciprocating engines the variable turning effort reacts on the guides and causes the engine to rock about its base. In certain circumstances this type of vibration can be very troublesome, as will be explained later on.

(*e*) *Singing Propellers.* There are different theories as to the precise explanation of this phenomenon. It is known, however, that by following certain lines of design singing can be avoided with a fair amount of certainty.

In the second group the following forms occur:

(*a*) *Vertical Vibration of the Whole Ship.* Most commonly this is of the fundamental 2-node type, although 3 and 4 nodes have been met. It is associated with engine balance in the vertical plane, and may also be due to dynamic unbalance of revolving parts of rotary machines, or propeller reaction.

(*b*) *Horizontal Vibration of the Whole Ship.* With 2, 3 or 4 nodes, it is caused by unbalance of revolving parts or by propeller effects.

(*c*) *Torsional Vibration of the Hull.* This has occasionally been referred to in technical circles and could conceivably occur with one or more nodes, principally stimulated by reactions at the stern of the vessel, particularly with multi-screws.

(*d*) *Local Vibrations.* Flapping of masts and rigging, panting of sections of the decks, vibration of doors, railings, furnishings, etc., can all be caused by machinery effects, although occurring at places remote from the source of disturbance.

(*e*) *Noise.* The noise spectrum of the engines usually includes a great number of different frequencies of varying intensity, the sound being transmitted through the structure, or airborne, so that the predominant notes may be heard in many parts of the ship more or less intensely. The intake of air to an engine or the discharge of the exhaust gases may make special contributions to this.

HISTORICAL SURVEY

Since the scope of the subject is very wide it is not surprising that from the time it became a major question it took over three-quarters of a century to identify the types, establish reasons for their occurrence, and devise means for dealing with the problems involved. It was undoubtedly fortuitous that in the beginnings of power-driven ships the combination and proportions adopted were unfavorable in general to the stimulation of vibration. The first commercially successful steamer was the *Comet* of 1812 and it was a small wooden paddle-driven vessel with a very slow running engine, an arrangement unlikely to be troublesome unless possibly due to the finish of the engine suffering from the inferior quality of machine tools available in those times. Iron hulls, more prone to vibration, came into use for sea-going vessels about 1832 and were supplanted by steel ships in 1870. It has been recorded that in some of the early paddle ships, although the speed of the engine was low the period of the floats striking the water coinciding with a natural frequency of the hull caused vibration. Even in modern paddle-driven river steamers forced longitudinal vibration of the ship, due to the variable turning effort of the engine, can be felt but not to an unpleasant extent.

As the size and speed of ships increased, associated with more powerful and faster running engines—which for a long period were exclusively of the reciprocating type—the relative proportions reached the zone where the possibilities of critical vibration occurring were greatly increased. Following successful experiments in 1837, screw propellers began to be adopted and there was a period during which the prevailing low-speed engines were fitted with increasing gear to suit a faster running propeller before higher-speed vertical direct-coupled reciprocating engines came into vogue, bringing the machinery into the range of vibratory conditions. Thus by 1884 much experience had accumulated leading Otto Schlick, who appears to have been the first to tackle the problem seriously, to assert in a paper[1] to the Institution of Naval Architects that "All steamers without exception shake to a more or less degree when the engines are in motion. This phenomenon is so natural that in most cases little or no attention is paid to it." Schlick then advanced the explanation that the vibrations were caused by the pressure of the reciprocating or revolving masses or the sideward pressure of the propeller. He was aware of the possibility of changing the conditions by altering the number of propeller blades, or moving the screw round relative to the engine cranks. In the latter event his supposition was that the center of gravity of the propeller was off center and a position could be found where the centrifugal force due to this balanced the engine effect. The influence of cargo disposition had been no-

ticed and strengthening of the hull after construction was dismissed as being ineffectively small in proportion to the total stiffness. A rough method of measuring vertical vibration on deck was illustrated which, although very crude, forms the basic principle in some modern instruments. Eight years later, in 1892, A. F. Yarrow in a paper[2] to the same Institution established that ship vibration was mainly due to unbalanced inertia forces in the engines and showed by a model how to counteract these by the use of bob weights. Engines were run coupled and uncoupled in a ship to show that it made no difference to the vibration. In a particular instance a maximum magnitude of vibration of $^{27}/_{64}$ in. measured at the stern of a torpedo boat was reduced to $^{7}/_{64}$ in. with the bob weights. An instrument called a Vibrometer for accurately recording the vibrations was exhibited.

In 1893 Otto Schlick read a second paper[3] describing his instrument called a Pallograph, for registering vertical and horizontal vibrations in ships, thus considerably facilitating analysis. At the same time the reason for periodic surging of vibration on twin-screw vessels was dealt with, the effect in the vertical direction varying from maximum to minimum as the relationship between the two crankshafts changed slowly from in-phase to anti-phase due to a slight difference in speed, and vice versa for the horizontal direction. The tendency for twin-screws to cause torsional vibration of the hull was commented on.

In the following year a further paper[4] by Schlick marked the publication of his well-known empirical formula for calculating the approximate fundamental natural frequency of vertical vibration of a ship. By means of an interesting model, Schlick demonstrated the influence of the relative position of the engine in the ship on the vibration produced—this being the first time attention had been drawn to this very important factor—and postulated the principles to observe according to the existence of unbalanced forces or unbalanced couples. Comments on unfavorable arrangements of 3-crank triple-expansion engines in tankers were made and 4-crank engines with particular crank angles balanced according to a given method were advocated. This paved the way for the introduction of the famous Yarrow-Schlick-Tweedy method of balancing steam reciprocating engines for ships, which revolutionized the harmonious running of the machinery. Schlick's activities stimulated so much interest in the matter that only a year later, in 1895, a group of three more papers on the subject was presented to the Institution of Naval Architects.[5] The first, by Schlick, was concerned with vibrations of higher order than the fundamental 2-node modes, these having been identified on certain types of ship. The free natural vibration characteristics of uniform rods had previously been established and Schlick stressed the importance of carrying out experiments on ships to locate nodal positions and obtain frequency ratios to compare with uniform bars. Torsional hull vibrations, not of large amplitude but known to affect humans disagreeably and to have destructive effects on the hull, had been experienced, and a type of formula for calculating their period was proposed.

The principal feature of the second paper,[6] by Mallock, was the description of a method of determining the period of vibration of a loaded ship by means of a wooden plank model, any section of which was proportionately as stiff as the ship at the corresponding position, loaded with weights equivalent to the cargo plus hull. The model was supported horizontally on two or three rollers suspended by crossed belts from similar rollers on overhead rails, and vibrated electro-magnetically. Arrangements were made to record the time and magnitude of vibration, the rollers being

adjusted to the positions giving maximum effect. The results showed that the nodes were farther from the ends than in a uniform bar, and the effect of changing cargo disposition was also examined on the model.

The third paper[7] in the group, by Robinson and Riall Sankey, was an attempt to profit by some experience with stationary engines by suggesting the adoption of two similiar 3-crank engines coupled in line for marine work as ensuring freedom from vibration effects, and for apparently the first time on record the influence of the obliquity of the connecting rod on the inertia forces, previously ignored as negligible, was taken into account.

In the discussion on these three papers R. E. Froude made the significant comment—"Of course we have often been obliged to put up with vibration. We are used to vibration in ships and when it was regarded as pretty nearly incurable we were ready to put up with it." This indicates that shipbuilders were going through some troublous times at that period. A fresh point mentioned was the effect of crosshead side thrusts creating a tendency for the engine to vibrate angularly about the center of the crankshaft.

Six years later Schlick made a further valuable contribution[8] to knowledge of the subject by giving an account of experiences on the trials of the *Deutschland*, a 662-ft. vessel with well-balanced twin-screw reciprocating engines totaling 37,000 s.h.p., the fundamental natural frequency of vertical vibration being 67 per minute. From pallograph records the maximum stern amplitude was 3/16 in. but did not occur when the two engines were in phase for the theoretically maximum effect, so Schlick advanced evidence to show that this was due to the propellers. At this time Thornycroft introduced the idea that want of uniformity in the velocity of the current following the ship could produce vibration. Also in the same year, 1901, Prof. Dalby published[9] his classical general solution for determining the unbalanced forces and couples in multi-cylinder reciprocating engines, taking into account the effect of the connecting rod.

It had by now become obvious that a reliable way of calculating the natural vibration periods of a ship was highly desirable and it was recognized that one of the main factors governing this was the effective modulus of elasticity of the material contributing to the strength of the hull, since the nominal value of 13,000 tons per sq. in. for steel could not be used for reckoning the stiffness in the principal directions of a large hollow complex structure having relatively thin members, as the distribution of load over any cross-section may depart from the usual simple beam theory, being influenced by buckling of plates, riveting, etc. On this factor Sir John Biles in 1905 described[10] some experiments made on H.M.T.B.D. *Wolf*, which was arranged to rest on two cradles in dock and be subjected to hogging and sagging with different levels of water, measuring deflections and strains from which the apparent values of E could be determined. The results do not appear to have given much satisfaction as they varied from 9300 to 16,000 tons per sq. in., throwing suspicion on the reliability of the tests.

In the same connection Prof. J. B. Henderson in 1909 gave an account[11] of experiments made on dynamic models consisting of flat steel bars resting on supports at the nodes and set in vibration electro-magnetically while the frequency was measured stroboscopically. First with a uniform bar it was demonstrated that nodal positions and frequencies tallied with theory. Then models of varying width and weights attached to simulate the stiffness and loading conditions on H.M.S. *Pathfinder*, for which vessel the natural fre-

quency was already known, were tried. From the results it was deduced that the effective value of E for the ship was only 70 percent of the value for plain bars. Similarly a model based on the *Lusitania* gave a ratio of 76 percent. In the light of present-day knowledge these experiments would be discounted.

Schlick's final contribution[12] to the subject was in 1911, when he reviewed the progress since his first paper 27 years earlier. He claimed that with 4-crank engines balanced on the Yarrow-Schlick-Tweedy system vibrations of engine frequency never occurred, and with balanced triple-expansion engines they could also be avoided, depending on the positional relationship of the engine to the nodes. Weak tremors remaining, formerly masked by the violence of the main vibrations, were attributed to the effects of the wake streams, particularly with inclined shafts, or to blade pitch errors in the propellers. In 1912 Gümbel[13] gave a method of calculating the torsional frequency of the hull by substituting for it an equivalent loaded elastic shaft, both torsional stiffness and the intensity of loading being proportional to the corresponding features of hull and cargo along the length, such system being amenable to mathematical treatment. Difficulties in application included doubt about the proportion of the structure to include in the stiffness and the effective modulus of rigidity. Nine years later Robb[14] referred to previous efforts to determine the effective modulus of elasticity and suggested that to a large extent the apparent discrepancies were due to neglecting the deflection arising through shear, a point already well established in the simple beam theory, and this view found general support.

About this time troubles attributed to torsional vibrations were commonly being experienced in the transmission of the newly-introduced forms of double-reduction geared turbine propelling machinery and a timely paper[15] on the subject was read by Walker and Cook, which showed that these were due to the varying resistance of the propeller blades in cutting through the non-uniform wake stream, and could be dealt with by suitable design. Interest in more accurate methods of calculating vibration periods for ships was increasing and in 1924 Nicholls[16] put one forward for obtaining 2- and 3-node vertical and horizontal frequencies allowing for bending and shear, with some attempt to account for the effect of water vibrating with the ship, the deflected form being assumed to be similar to that for a uniform bar vibrating in the same mode. Dynamic models were used and the results compared with records from an actual ship. Nicholls also improved on Gümbel's method of calculating torsional periods of the hull by making the masses distributed instead of concentrated and dispensing with the calculus.

In 1926 an article[17] by Pavlenko was published explaining a new method of calculating 2-, 3- or 4-node ship periods by making certain assumptions about the variation of section inertia and loading and using tables of appropriate coefficients based on the deflected form being as for a uniform beam. It was claimed that this gave closer results than any other known method but factors were neglected which are now known to be quite important.

The first paper[18] on the subject given to this Institution was by Cole in 1928. It introduced a new method for the calculation of the flexural periods of the hull using a complex formula derived by equating the work done in deflecting the ship to the sum of the bending and shear strain energies, accepting that the curve of deflection was similar to that for a uniform bar vibrating in the same mode, with an approximate correction to give proper end conditions, taking an equivalent parabolic form of loading, and using coefficients for different types

of vessel to allow for variation of section inertia. There was also an allowance for added water according to empirical rules. To shorten the work, tables of values for ranges of some of the differential coefficients and integrals required were included. In view of the broad assumptions and factors ignored the treatment could not be regarded as complete or a great advance on the simple approximate short-cut formulae available at the time, although in the examples given three out of four calculated periods agreed closely with observations. Some consideration was also given to longitudinal and torsional hull vibrations.

A paper[19] by Inglis in 1929 dealt with the development of mathematical methods of calculating the natural frequencies and modes of vibration in beams of non-uniform mass and section but it was hardly suitable for direct application to ships. Around this period some unpleasant experiences had arisen in Diesel-driven vessels and in 1930 Sulzer read a paper[20] "Causes and Prevention of Vibration in Motorships." This paper was mainly concerned with torsional vibrations in the engine and transmission shafting, and how these could be restricted by changes in shaft diameters, crank angles, and flywheel masses, but some examples of other types of vibration were also included. Instances where high-speed auxiliary engines caused local vibrations were mentioned. Attention was drawn to transverse vibration of engines causing shaking of the foundation and hull. The opinion was expressed that engineers would be satisfied if the hull frequencies could be estimated within 5 percent of the actual values.

One of the most useful contributions to the subject was the paper[21] by Lockwood Taylor in 1930. This dealt with a method of estimating the probable magnitude of ship vibration allowing for damping and for the relative position of the engine in the ship, comparisons between observed and calculated deflections for a number of ships under different loaded conditions being given. In a paper[23] in 1933 Inglis gave the germ of the idea for minimizing vibrations by the application of a small auxiliary mass attached to the affected part. Five years later Constantini produced a paper[25] showing how to calculate ship periods by using more suitable and reliable coefficients in the empirical formula, taking into account the entrained water, the distribution of mass and inertia, the effective elastic modulus and inertia, the damping factor, and rotational kinetic energy. Points discussed included uneven stress distribution, imperfect workmanship, temperature effects, effectiveness of longitudinal frames in tankers and of light superstructures on passenger vessels, wood sheathings and deck compositions, etc. Diagrams obtained on a Geiger vibrograph showing vibrations of various types on several vessels were given, including one showing the effect of a damaged propeller blade. A type of vibration difficult to eliminate was referred to, namely, that arising from the propeller, and a device for coping with this, the Loser Neutralizer, was described and illustrated. This contrivance, which seemed a practical outcome of Inglis' 1933 suggestion, had been fitted in a single-screw motorship with satisfactory results, particulars of the performance with and without the neutralizer in action being given. When installed on a large passenger liner, however, the effect was not so complete as the neutralizer could not be located near enough to the source of disturbance. The principle of this fitting is that by adjusting the mass of water contained within it its frequency can be fixed just below that of the disturbing impulse and it therefore moves in anti-phase to the latter, reducing the movement to an extent depending on its size in relation to the impulse. In a ship loaded to 11,700 tons the neutralizer

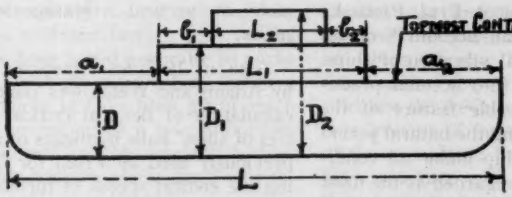


Fig. 1.—Ship vibration frequencies.[27]

I.—*Equivalent depth of ship*

$$D_E = \sqrt[3]{D^3(a_1+a_2)/L + D_1^3(b_1+b_2)/L + \ldots D_2^3 L_2/L}.$$

II.—*Effect of vibrating water*

Δ and Δ_1 are displacements, tons, actual and allowing for water

B = Breadth and d = draught in ft.

$$\Delta_1 = \Delta[(B/3d)+1{\cdot}2].$$

III.—*Frequency formula*

N = Vibs. per min., vertical 2-node, β = constant from graph.

$$N = \beta\sqrt{BD_E^3/\Delta_1 L^3}.$$

Deduced from the graphs $= 46{,}200\sqrt{BD_E^3/\Delta_1 L^3} + 27$ for passenger and cargo ships

$= 51{,}200\sqrt{BD_E^3/\Delta_1 L^3} + 30$ for tankers.

IV.—*Vertical 3-node vibration*

Ratio 3-node freq.: 2-node freq. = 2·28–2·53 : 1 (measured on 3 ships).
For uniform beam = 2·76 : 1 (theoretical).
For double wedge-shaped beam of uniform depth = 2·26 : 1 (theoretical).

V.—*Horizontal 2-node vibration*

Approx. 1·37 × vert. 2-node freq. at 80 per cent. full displacement
1·25 × vert. 2-node freq. at 57 per cent. full displacement
say $(0{\cdot}3x+1{\cdot}1)$ × vert. 2-node freq., where x = fractional displacement.

weighed 12 tons and reduced the vibration 94 percent. From the discussion on this paper it was evident that severe vibrations were still very prevalent on ships.

For some time prior to and following this, eminent British authorities—notably Todd[22] and Burrill[24]—devoted much attention to the problem, principally in the direction of rendering the simpler methods of calculating ship periods more closely accurate. The latest 1948 version[27] of Dr. Todd's method is very simple to apply but is likely to give errors up to 10 percent or more unless a greater range of coefficients to cover all designs of ships built is amassed from actual experience (Fig. 1).

Within comparatively recent times research workers in other countries have made an extensive study of the subject and carried out some very thorough investigations. As a result of several years of theoretical and experimental

effort on the Continent, Prof. Prohaska in 1947 published an account[26] of his work on the vertical vibration of ships. It appears to take into account practically every conceivable feature of the problem of calculating the natural period of vibration of a ship under all conditions and might be regarded as the most complete solution so far published. Briefly the work consisted of first calculating the period for a number of different types of vessels of varying form with many different distributions of loading and inertia, this being carried out by the known lengthy process. The results were related to form parameters, with methods of allowing for shear, superstructure, additional mass of water, draught, trim, and depth of water, so that for any of the usual types of vessel the period could be calculated by means of comparatively simple formulae used in conjunction with corrections obtained from graphs or tables. (Appendix I.)

The effects of draught, depth and mass of water, were verified experimentally with models of different forms tested in a tank at several draughts. For 15 ships of various types the values obtained by the calculations were exceptionally close to the actual observed frequencies. The ground covered, however, only applied to the fundamental state of vertical vibration with two nodes.

Also in 1947 an American report[28] by Adams and Welch was issued on the calculation of flexural critical frequencies of ships' hulls by means of a method previously used by Prohl for determining the critical speeds of turbine rotors, etc. The method, which is applicable to the fundamental and higher modes, implies the use of a punched-card machine, a knowledge of the distribution of mass and section inertia of the hull, and of the mass of water affected. It appears, however, to ignore some important factors.

A recent paper[32] by Jaeger repeated or confirmed much of Prohaska's results, with extension to the horizontal vibrations and higher modes, torsional hull vibrations, and consideration of the assessment of the nuisance value of vibrations. Apart from the question of hull frequency calculations, there have been during the past ten years or so a number of articles or papers on other branches of the subject, including instrumentation,[37, 39] vibration diagnosis in geared turbines,[30] torsional axial, and lateral vibrations in the transmission system,[34] measurement and analysis of torsional vibrations in the engine system,[42] transverse vibration of marine Diesel engines,[36] and special methods of performing vibration calculations.[35, 41]

PRESENT STATE OF KNOWLEDGE

HULL VIBRATION PERIODS

Vertical Vibration. By means of comparatively simply applied modern formulae the natural frequency of 2-node vertical vibration at a given displacement can be estimated from the basic data usually available in the preliminary design stage. The degree of error may amount to as much as 10 percent, but this margin could be reduced as more information becomes available about actual observed frequencies for representative types of vessel from which more appropriate values of the constants could be determined. For closer estimation, and to allow for more varying conditions, the method developed by Prof. Prohaska[26] can be utilized.

The long-term full integral methods that have been established for calculat-

ing this frequency, taking into account all known relevant factors, give results that have been found generally to agree very well with observed values, but they do not appear to have been given much attention excepting by a few authorities on the subject and by research bodies, presumably because of the great amount of work involved in the task.

As regards the higher modes of vibration, that is, with 3, 4, or more nodes, no reliable approximations have been devised and the integral method used for the fundamental mode is inapplicable. There are, however, some other devious ways of proceeding, including solutions by the energy method and by the use of basic functions,[88] which with proper allowances show some promise of satisfaction, and continuing research on these may lead to something tangible in the near future. As the number of nodes increases the proportional influences of shear and rotational inertia on the deflected form become greater, and the effect of these factors is difficult to assess precisely. Observations have shown that the ratios between the frequencies in the several modes of vibration, although fixed quantities for uniform free-free bars, vary widely in practice, even in the same ship under different conditions of loading. Since the higher modes may be stimulated by effects from numerous sources, for example, the main or auxiliary machinery and propellers, and generally are more unpleasant to endure than low periodicity disturbances, it is considered desirable to have a reasonably close idea of their frequencies before the design is too far advanced, thus the pursuit of further knowledge on this question should be encouraged.

Horizontal Vibration. As a first approximation, the natural frequency of the fundamental horizontal vibration may be obtained by multiplying the corresponding vertical frequency by a factor depending on the percentage of full displacement and type of vessel. Examination of published figures recorded on ships shows, however, that this ratio varies very widely. Attempts to calculate the frequency accurately by the long process of integration, etc., similar to those explored for vertical vibration have not met with much success and this seems to be principally because of the difficulty of properly allocating the amount and distribution of the mass of water associated with the vibration. Some investigation is, however, being carried out in that direction. Meantime it seems that only the collection of much data from actual observations on a large number of different vessels under different conditions can afford an adequate means of dealing with this problem, which does need consideration because horizontal 2-node vibration is commonly met with and reputed to be more uncomfortable than the vertical vibration.

The situation in regard to the frequencies of the higher modes of horizontal vibration is even more obscure than for the corresponding vertical types but fortunately horizontal disturbing impulses tending to stimulate these are less likely to arise, are lower in magnitude, and easier to control.

Torsional Hull Vibrations. Although these have frequently been referred to in papers, approximate formulae being quoted and methods of working out the frequency described, little factual evidence has been published on the subject. Only the fundamental mode, that is, with a node approximately midships, has been mentioned, and the frequencies appear to be at least several times the vertical 2-node transverse frequency. In calculations, one of the stumbling blocks is the difficulty of assessing the effective torsional stiffness of the structure. Twin- or multi-screw vessels appear to be those most likely to be affected by this type of vibration.

Longitudinal Hull Vibrations. The possibility of these arising is very re-

mote but should not be entirely ruled out for futuristic ships of extreme dimensions. Frequencies have not been worked out in detail but are believed to be so high as to put the conditions outside the range of appreciable synchronous effects.

Torsional Vibrations in the Machinery. All forms of propelling machinery are subject to the possibility of torsional vibrations being excited within the transmission system, but some are more prone to this than others. Neglecting the mass of the connecting shafts there are as many possible modes of vibration as there are shafts in the system but usually in practice only the first or second require consideration, although the third and higher modes sometimes have to be reckoned with.

Methods of determining the torsional vibration characteristics when vibrating in any of these modes have been established on firm foundations and conditions are capable of being predicted with sufficient accuracy for most purposes, within the limits of error of $\pm 2\frac{1}{2}$ percent for 1-node and ± 5 percent for 2-node frequencies. There are various ways of working out the natural frequencies of torsional oscillation, the most practical being the simply applied trial-and-error tabular process attributed originally to Holzer, and this is facilitated if a suitable machine is available and if advantage is taken of preliminary approximations. If in special instances a deeper investigation is envisaged, for example, involving the effect of adjustment of shafts and masses, then some of the newer methods that have been more recently introduced are applicable, for example, such as the use of tuning curves[41] and the conversion of the problem into its electrical analogy,[35] both of which require more mathematical skill and concentration than the first method.

The correct assessment of the basic data is of primary importance and mention should be made of some of the doubtful factors. It is a common practice to add 25 percent to the propeller inertia to allow for water carried round, regardless of speed, immersion, and shape and flexibility of the blades, which does not seem logical, and some authorities have suggested more rational treatment, associating the addition, for example, with the solid of revolution swept out by the blade profile,[20] or the pitch ratio.[29] However, quite a large error in the allowance will in many instances only have a small influence on the frequency.

Propeller inertias in air calculated from approximate formulae are subject to wide discrepancies (see Appendix II), and for final calculations it is advisable to obtain the inertia by graphical integration of the blade sections. The figures for inertias of important elements such as flywheels, dampers, rotors, and gear-wheels, can be checked experimentally by swinging or rolling on knife edges.

In motorships, the torsional stiffness of the crankshaft has a decided influence on the second and higher modes of vibration, and experience shows that it can be estimated sufficiently closely to agree with observations by adaptations of the general empirical formulae that have been published and amplified in recent times, verified in some instances by actual tests. These include the Carter formula, the British Internal Combustion Engine Research Association rule,[33] and Zimanenko's method.[33] By the use of such means, for a series of direct-acting opposed-piston engines the crankshaft stiffness ratio for different sizes ranges from 1.02 to 1.21, this being the ratio of the equivalent length of plain shaft to the actual length. Similar engines made by different manufacturers may not show exactly the same characteristics, as the methods employed, bearing clearances, alignment, etc., vary, and these have an influence.

As regards the disturbing impulses which foment the vibrations, in turbine drives the principal effects arise from the varying propeller resistance, those due to gear-cutting errors and alignment faults being of unpredictable amount, although very high standards are now achieved in that respect. In motorships, while variation in propeller resistance is likewise a factor, the predominating influence is the varying turning effort of the engine. For the principal proprietary types of oil engine the harmonic component of the turning effort due to gas pressure in the cylinder have been determined over the output range. In this connection it should be stated that the indicator cards used as a basis must be perfect, preferably obtained with a geometrically correct direct-driven rig or by means of the Farnborough indicator, the cam-type gear being subject to too great a possibility of error. For the lower orders these have to be combined with the harmonics of the inertia forces that can be calculated, taking into account the ratio of the connecting rod to the crank.

For the major modes of vibration, if both propeller and engine sources of excitation are relatively important, then the phase relationship between the blades and cranks can have a substantial influence on the value of the resultant vibratory torque, but generally little attention is given to this point, the setting adopted being an arbitrary one having no particular reference to calculated vibration characteristics. To illustrate this, a recent paper[34] deals with the hypothetical case of a midship installation of triple-expansion engines driving a 3-bladed propeller, the conclusion arrived at being that with correct phasing the vibratory torque reached a value 77 percent of the mean torque, whereas in the worst phasing position this rose to 127 percent. At the same time it was pointed out that the effect on the propulsive performance of the propeller of the torsional vibration in conjunction with the wake form would also have to be considered in fixing the relative angle.

Methods of calculating the probable vibration stresses at synchronous speeds vary according to the mode. For 1-node vibration where the propeller is the principal source of damping, the magnitude of this can be approximated from known characteristics of the screw or prototype model for use in expression giving the stress.[42] With 2 or more nodes, the propeller amplitude is relatively small, therefore only engine damping and shaft hysteresis need be considered, some taking an over-all engine damping factor to cover these and others basing on shaft hysteresis laws with a suitable coefficient determined from experience to allow for other factors. A recent classification society recommendation is that certain stated limits of vibration stress in the main driving parts should not be exceeded, distinguishing between those allowed for continuous running or for transient conditions, taking into account the relationship of the critical speed to the service speed and the size of the shaft. No guidance, however, is given as to the methods to be employed for ensuring this by previous calculation. This is no doubt a necessary step. Nevertheless some installations produced in the past—before much attention was given to the subject—were liable to decidedly more severe conditions than would now be accepted and some of these are still running without experiencing failure due to torsional vibration. A certain amount of control is possible by adjusting shaft stiffness or the size and disposition of flywheel masses. Fig. 2 shows the effect of varying the size of the transmission shafting on the 1-node conditions in a tanker installation. As the diameter is increased to reduce the vibration stress to continuous running limits the critical speed draws nearer to the service speed, which might be considered more objectionable than having a barred range lower down the scale,

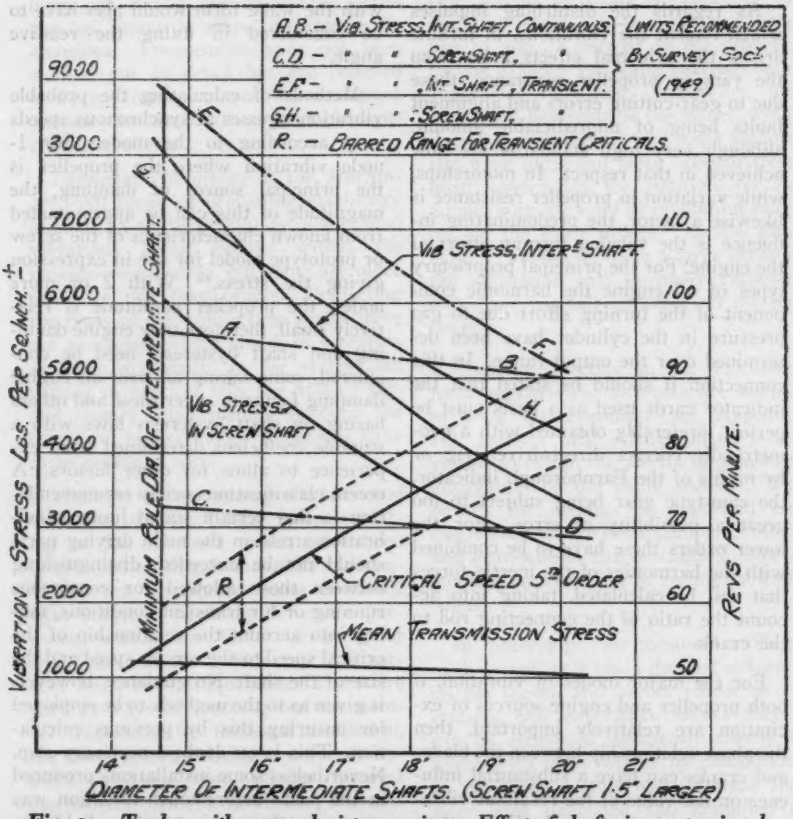


Fig. 2. ***Tanker with opposed-piston engine. Effect of shaft size on torsional vibration stress and critical speed.***

especially if the astern power is only about half the ahead power. In this particular instance, the size of shaft necessary to put the critical speed well above the service speed would be prohibitive.

As regards the actual measurement of the amount of torsional vibration, mechanical instruments such as the Geiger and Cambridge torsiographs are frequently used to produce permanent records for subsequent analysis. Although subject to certain limitations they are favored for ease of application and operation and can yield reliable results. Modern types of electronic apparatus can also be applied, including the wave analyzer, but on account of their complex nature their use is best confined to specialists until marine engineers become accustomed to the apparatus and are convinced of its reliability. The point of application of the instrument requires judicious selection according to the mode of vibration and should not be

where stiffness varies. If applied at only one spot, the correctness of the estimated elastic form of the system has to be assumed in calculating the vibration stresses, and depending on the amount of damping and the proportions of the system comparatively large errors could arise if due allowances are not made. The most reliable course to adopt, which has been used for example for 1-node vibrations in intermediate shafting, is to apply two torsiographs as far apart as practicable without a node or mass between and take simultaneous records yielding the vibration twist by difference and hence the stress by direct calculation.

When the vibration stresses exceed the limit regarded as safe for continuous running it is usual to mark barred ranges on the tachometer and to exhibit a warning plate. The normal conditions, however, could be upset by special circumstances such as uneven distribution of power in the cylinders of reciprocating engines, the cutting out of one or more cylinders in emergency, tachometer belt slip, and damage to propeller or damper. It would, therefore, appear useful to have a simple device to give visible indication of the incidence of critical speeds, since these do not necessarily give external warning of their presence.

HULL VIBRATION EXCITED BY THE ENGINES

Turbine Machinery. Transverse vibration with high-speed rotating parts is caused when the mass centroid line of the revolving element is not coincident with the geometrical axis of the journals, centrifugal action setting up unbalanced revolving forces reacting on the bearings which in the general case are unequal and not at the same angle, the effect being equivalent to a single force and a couple. The frequency of these is usually well above the frequencies of the low orders of vertical or horizontal hull vibration, but the bearing pedestals, seatings, pipes and fittings directly attached, are liable to be seriously affected and to transmit vibration to any remote part of the vessel where the local natural frequency synchronizes with the speed at the source, a condition usually amenable to local stiffening. Due to slight machining errors, lack of homogeneity of the metal, and non-uniformity of attached parts, a 5-ton rotor for 3000 r.p.m., for example, might be out to the extent of anything from 5 to 50 in.-oz. per ton weight, corresponding to unbalanced forces at speed of between 400 and 4000 lb. After adjustment in a modern dynamic balancing machine it is possible to reduce the cold error to the region of 1 in.-oz. per ton, the force then being only 80 lb., which should ensure smooth running unless in special circumstances such as thermal distortion due to high temperature or uneven heating, damage to blading, rubbing in packings, overspeeding, priming of steam, uneven deposits, etc. By means of a special type of balancing machine or with electronic equipment, it is possible to rectify the effects of mishaps or deterioration by re-balancing the part on board should that course be necessary, although at reduced sensitivity compared with a normal machine ashore. In low-speed parts like large main gearwheels, which with cast iron centers may be as much as 50 lb. out near the rim when finish machined, sufficient to create an unbalanced force of the order of 1000 lb. at speed, static balancing on knife edges is usually quite sufficient to ensure that no forces capable of exciting appreciable vibration are set up.

Reciprocating Engines. With these the tendency to produce vertical or horizontal vibration in the ship is mainly dependent on the characteristics of the unbalanced inertia forces arising from

reciprocating or revolving parts, on the position of the engine in the ship, and on the speed in relation to the natural periods of the vessel. Revolving effects occur at engine speed which limits any consequent horizontal vibration to frequencies at or near that rate. With usual proportions harmonics of the 1st and 2nd order are the only ones generally of any significance arising from the inertia of the reciprocating parts, so that in the vertical plane synchronism would take place if the vibrations of the ship per minute are equal to or twice the revolutions of the engine per minute. Usually the aim is to balance engines as completely as possible, but even engines that are considered to be very well balanced are not necessarily immune from vibratory influences, because the balance criterion is an over-all group feature, and in long engines the difference in amplitude of vibration throughout the length, taking into account the deflected form of the hull, can have a decided effect, the stiffness of the frame having some bearing on this. A near approach to the ideal is a well-known and predominant type of opposed-piston engine in which the revolving and primary reciprocating forces for the upper and lower drives at each individual cylinder balance each other, leaving only 2nd-order forces, which are of smaller magnitude, to deal with by group balance. Such engines can be confidently installed in any of the usual positions favored in a ship without fear of causing vertical or horizontal vibration provided that the balance of attached pumps is adequately covered.

It is not generally appreciated that engines very much out of balance according to generally accepted ideas can, with suitable disposition, run at synchronous speeds without setting up appreciable vibration. To advance the study of ship vibration by means of apparatus applied directly on board the Lloyd's exciter[29] was introduced. This can be used to subject the hull to periodic forces of known magnitude and frequency but its scope is restricted by the size and weight, so that systematic exploration of the effects of application at different positions along the profile of the ship is difficult to arrange.

Transverse Engine Vibration. In reciprocating engines the reaction to the varying turning effort tends to cause the frame to vibrate transversely about its base, the guide pressure forming one arm of the couple acting. The combination of engine and double bottom, both possessing some degree of flexibility, has a natural frequency for this type of vibration, and when this synchronizes with the frequency of any of the harmonic components of the overturning effort amplitudes may reach proportions which not only cause serious inconvenience in the engine room but set up secondary effects in every part of the ship where elements of the same frequency exist. This engine condition, which is loosely termed "tuning-fork" vibration, is one of the most difficult to eliminate. It is rather surprising, therefore, that very little factual matter about it has been published. Generally only the major harmonics, which act in consort, need to be considered; thus for example in a single-acting two-stroke engine the speed should preferably be kept clear of the natural frequency divided by the number of cylinders. This is not always possible when the number of cylinders is large but against that the higher harmonics are smaller in magnitude and if the seats and double bottom are well stiffened the amplitude can be limited to acceptable amounts, with the critical speed in a convenient relation to service speed. However, it seems that large engines with numerous cylinders, in addition to vibrating transversely in the parallel mode, can also respond in a skewed fashion, the upper part bending like a double cantilever fixed in the center, or as a bow with 2 nodes, in which event the resultant effect of the harmonics would require closer study,

although with these more complex modes the reactions on the ship would not appear to be so serious. Some information about this aspect is given in a recently published paper[36] by Ijsselmuiden. Tuning-fork vibration in twin-screw ships can be alleviated by putting ties between the engines, and this raises the natural frequency. In single-screw ships the fitting of ties between the top of the engine and the sides of the engine room casing is sometimes proposed to reduce transverse movement but for several potent reasons few practical engineers would agree to this, especially in large vessels.

Propeller Effects. If the center of mass of the propeller is displaced from the axis of rotation or the blades are not exactly similar in pitch and spacing, then at speed the supporting bearing is subjected to varying reaction occurring at shaft frequency which can excite vertical or horizontal hull vibration. This was a common occurrence in the earlier days of screw propulsion but now that most modern working propellers are made by precision methods, finished all over and statically balanced, this feature can be considered eliminated unless the screw becomes damaged. Even with perfectly made propellers, however, there can be exciting impulses set up at blade frequency or multiples thereof due to wake variation, hydrodynamic pressure fluctuation on the submerged hull in proximity to the sweep of the screw, or to the passage of the blades past bossings, struts, stern frame, and rudder, the extent of these effects being very difficult to assess. Some data have accumulated on wake distribution, the variation of which can be minimized by adopting suitable lines for the after body, and as regards the other items adequate clearances in the way of the blade tips have to be allowed. Unpleasant results from this type of action are by no means uncommon and the obvious but expensive cure is to replace the screws by others with a different number of blades and thus alter the frequency.

When the ship is vibrating the amplitude at the extreme ends is several times the greatest amount elsewhere, so that disturbing impulses due to propeller action have a much more powerful stimulating effect than forces of the same magnitude occurring in machinery situated closer to midships. In recent years there has been an epidemic of screw-shaft casualties in certain classes of ship, failures occurring mainly in the tapered end, indicating bending fatigue principally, and a great amount of investigation into this has been carried out. One authority[34] asserts that the main cause of the trouble is local lateral whirling of the propeller on the overhanging shaft and advances a theory confirmed by model experiments in support of this idea. Due to the thrust variation on each blade as it cuts through the non-uniform wake the shaft is subjected to a fluctuating moment causing it to bend in a fore-and-aft plane as it rotates, the number of reversals of stress per revolution being dependent on the number of blades. When resonance occurs the propeller whirls with the shaft and the static stress due to the eccentric thrust load is considerably magnified, so that in conjunction with other effects high stresses leading to ultimate failure are set up. Ship vibration will also accompany the state if the frequency is near a hull frequency. This theory appears to be worthy of close study.

Local Vibrations. These are liable to occur in any part of a ship and in such a complicated structure it is impossible to guarantee complete freedom before completion. They are due to local frequencies synchronizing with disturbing impulses in some section of the running machinery. Generally it is possible to eliminate them by extra stiffening of the part, particularly if the auxiliary concerned is a constant-speed machine. There are instances, however, where it

has been found more satisfactory to deal with the source of the disturbance direct, for example, by improving the balancing or mounting arrangements. In motor ships vibration of the engine room casing and surrounding accommodation is a common source of annoyance. It can be caused by the exhaust impulses, and this should be borne in mind when designing the exhaust pipe lead and anchoring arrangements.

With the opposed-piston engines vibration of the engine room casing doors and in adjacent passages has sometimes been experienced. In a particular example, the entrance door on one side of the casing opened into a long passage closed at one end. Near full speed the panels of the accommodation along the outside of the passage vibrated heavily when the door was open. The engine had 5 cylinders and a single crank-driven scavenge pump. Application of a reed vibrometer indicated that the frequency of the vibrations was five times engine speed. The upper pistons caused air movements at the same rate and rough calculation showed that the resonance in the casing air space was unlikely. Treating the passage way as an organ pipe with one open end, it appeared that the air impulses were synchronizing with a natural frequency of the air column and of the panels. The fitting of a double door with an air lock between ensures freedom from this type of action.

LONGITUDINAL VIBRATION OF THE SHAFTING SYSTEM

Instances of the above are rare in the mercantile marine but during the last 10 years or so experiences on high-powered naval vessels and a large passenger liner led to a thorough exploration of the principle involved so that for future guidance there is ample information on the subject available.[31] With this type of vibration the thrust variation is considerably magnified if resonance occurs, imposing heavy loads on the thrust block and seating, which should be designed accordingly and the most appropriate number of blades in the propeller chosen to suit the condition.

SINGING PROPELLERS

A comprehensive review of the accumulated information on this topic, together with some broad conclusions derived therefrom, is given in a recently published article[40] by Work.

NOISE AND VIBRATION SENSATION

The continuous metallic nature of the ship's structure is ideal for transmitting noise from the machinery to all parts, and air waves also act as a medium. A certain amount of practical investigation has been carried out as to the extent to which noise reaches the accommodation, the proportions that are respectively structure-borne or air-borne, and the effectiveness of sound insulating arrangements, and possibly in time this may lead to more uniformity of opinion on the subject. Noises from high-speed engines, particularly of the internal combustion type, are more objectionable than from slow-moving machinery, and it is a common experience to find that direct-coupled main propelling Diesel

engines are much quieter than the faster-running Diesel generators. Enclosing the latter in sound-proof cabinets is one method that has been formulated as a preventive. Another modern artifice is to support the complete unit on noise isolation mountings. The principles underlying the design and application of these fittings are now well-established and they are becoming more widely used. For the main propelling machinery, however, there are strong practical objections to the adoption of resilient mountings.

One of the difficulties in dealing with this matter is the absence of a simple and satisfactory method of assessing the strength of the noise without having to depend merely on the judgment of individuals. Too many bases of comparison are in vogue and the marine engineer is apt to be confused with the multiplicity of terms used in noise and vibration measurement, for example, decibels, phons, sones, pals and trems, so some clarification and standardization appears to be needed.

The silencing of the exhaust gases from internal combustion engines is a problem that demands attention, particularly in passenger ships. Numerous designs have been tried in the past but so far the ideal of maximum sound absorption with minimum resistance does not seem to have been sufficiently attained, and it seems surprising that there has been no organized research on this to establish basic principles. High back pressure is detrimental to the performance of the engines, especially those on the two-stroke cycle, and the silencer must be so proportioned that it gives sufficient acoustic efficiency without exceeding the allowable resistance. Waste-heat boilers generally serve as excellent silencers, also exhaust-gas turbines. Methods in use for silencing the air inlets to oil engines and turbochargers give satisfaction.

Like noise, the question of the nuisance value of mechanical vibration is a vexed one. Authorities have divergent views on the basis of measurement and Table I gives the factors to which the degree of discomfort has been considered proportional by different investigators.

TABLE I

Criterion	Expression
Amplitude	α
Velocity (max.)	$\alpha\omega$
Acceleration (max.)	$\alpha\omega^2$
Kinetic energy (max.)	$\alpha^2\omega^2$
Rate of change of kinetic energy (max.)	$\alpha^2\omega^3$

ω is the circular frequency of the vibration.

The velocity criterion appears reasonable from the point of view of the stresses set up, and suits some, convenient methods of instrumentation.[30] The acceleration criterion seems more suited to the idea of human reaction to the vibration; accelerometers can be used for recording. Data have been published on each of these conceptions and it seems desirable to adopt a common method of approach.

A point worth mentioning is that whereas vibrations in the machinery can produce dangerous stresses in the working parts, vibration stresses in the hull are generally small even though the amplitudes appear serious to the senses.

FUTURE TRENDS

Modern requirements in connection with engineering products have become so exacting that it is astonishing to find little or no reference to vibration characteristics in contract specifications. The fact of the matter is that this cannot be done in a definite manner unless and until proper standards of measurement are available. Another point is that usually no attempt is made to decide the

onus of responsibility as lying with the engineers or the shipbuilders, which could in some circumstances create an awkward situation. The subject is one which demands the closest possible collaboration between naval architects and engine designers, and there is a tendency for this to be neglected, particularly when another firm supplies the engines to a shipbuilder. In time complete knowledge of the subject may render it possible for manufacturers to work with confidence to rigid specifications. They will, however, need to be on guard against the ever-changing conditions associated with engineering progress, the introduction of new types of machinery or more advanced versions of existing forms; for example, gas turbines and supercharged two-cycle oil engines are possibilities.

Materials have an influence, and the use of aluminum in ship construction is being increasingly advocated. All-welded ships are becoming more numerous and in this connection experience has already shown some difference as compared with similar riveted ships, that is, less structural damping capacity.

Improvements are needed in regard to the simplicity, reliability and mobility of relevant apparatus, not only for research purposes but also to enable ordinary operators to diagnose vibration complaints with dispatch.

It is interesting to note that increasing attention is being given to the application of strain gauges to places hitherto considered inaccessible, for example propellers, which should prove helpful in vibration study.

A great amount of research has been carried out recently, and is continuing, on ship vibration problems, particularly in regard to frequencies. Possibly too much emphasis has been laid on engine effects, because if the position of nodes is known approximately, engines could be so designed that at synchronous speed no appreciable vibration resulted, and consequently the exact value of the frequency would be immaterial. This would simplify matters provided that effects arising at the propeller end could be rendered insignificant, and more attention to that aspect would be profitable.

Appendix I

Vertical Vibration of Ships

Prohaska's Formula [26]

$$N = [55.5\ R/\sqrt{q(1+e)}]\ \sqrt{I/[(\Delta/1000) \times (L/100)^3)}].$$

N = Number of 2-node vertical vibs. per min.
I = Midship section inertia about the neutral axis, ft.^{4}
Δ = Displacement in tons.
L = Length, ft.
q = Total mass distribution coefficient.
e = Ratio of mass of vibrating water to displacement.
r_1 = Correction factor for variation of section inertia.
r_2 = Correction factor for deflection due to shear.
r_3 = Correction factor for transverse contraction.
R = Product $r_1 r_2 r_3$.

Graphs are given to aid in assessing R, q and e.
e has to include water depth allowance, by graph.

Appendix II

Moment of Inertia of Propellers

A commonly used expression for weight is of the form:
where

$$W = K_1 D^3 (M.W.R.)(B.T.F.)$$

W = weight, tons.
D = Tip diameter, ft.
M.W.R. = Mean blade width ratio.
B.T.F. = Blade thickness fraction.

Likewise for inertia:

$$I = K_2 W D^2 \text{ ton ft.}^2$$

Data from a representative number of different solid 4-bladed propellers gave the following values for the coefficients:

K_1 = 0.178 to 0.228, variation 28 percent, for bronze.
0.169 to 0.194, variation 15 percent, for cast iron.
K_2 = 0.0368 to 0.0464, variation 26 percent, for bronze.
0.0416 to 0.0490, variation 18 percent, for cast iron.

Closer estimates can be obtained by combining these formulae, thus:

$$I = K_3 D^5 (M.W.R.)(B.T.F.)$$

where

$K_3 = K_1 \times K_2$
= 0.00786 to 0.00916 for bronze (16½ percent variation).
0.00727 to 0.00854 for cast iron (17½ percent variation).

These values of K_3 are the same propellers which gave the above values of K_1 and K_2.

APPENDIX III

DOUBLE REDUCTION GEARED TURBINES WITH FLEXIBLE SHAFTS

Frequency equation:

$$I_0 + I_1/\delta_1 + I_2/[\delta_2 - I_4/(\delta_4 I_2 + I_4)] + I_3/[\delta_3 - I_5/(\delta_5 I_3 + I_5)] = 0.$$

I_0 = Inertia of secondary gearwheel and pinions.

I_1 = Inertia of propeller.

I_2 & I_3 = Inertia of H.P. and L.P. primary gearwheels and pinions.

I_4 & I_5 = Inertia of H.P. and L.P. turbines.

$\delta_1 = 1 - (F/n_1)^2$, $\delta_2 = 1 - (F/n_2)^2$, etc.

n_1 = Frequency of propeller with gearing fixed.

n_2 & n_4 = Frequency of H.P. and L.P. primary gears on their shafts with the secondary gear fixed.

n_3 & n_5 = Frequency of H.P. and L.P. turbines on their shafts with the primary gears fixed.

F = Natural frequency of whole system (5 values in all).

The inertias are all referred to the low-speed shaft.

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THE WORTHINGTON-JUNKERS FREE PISTON AIR COMPRESSOR

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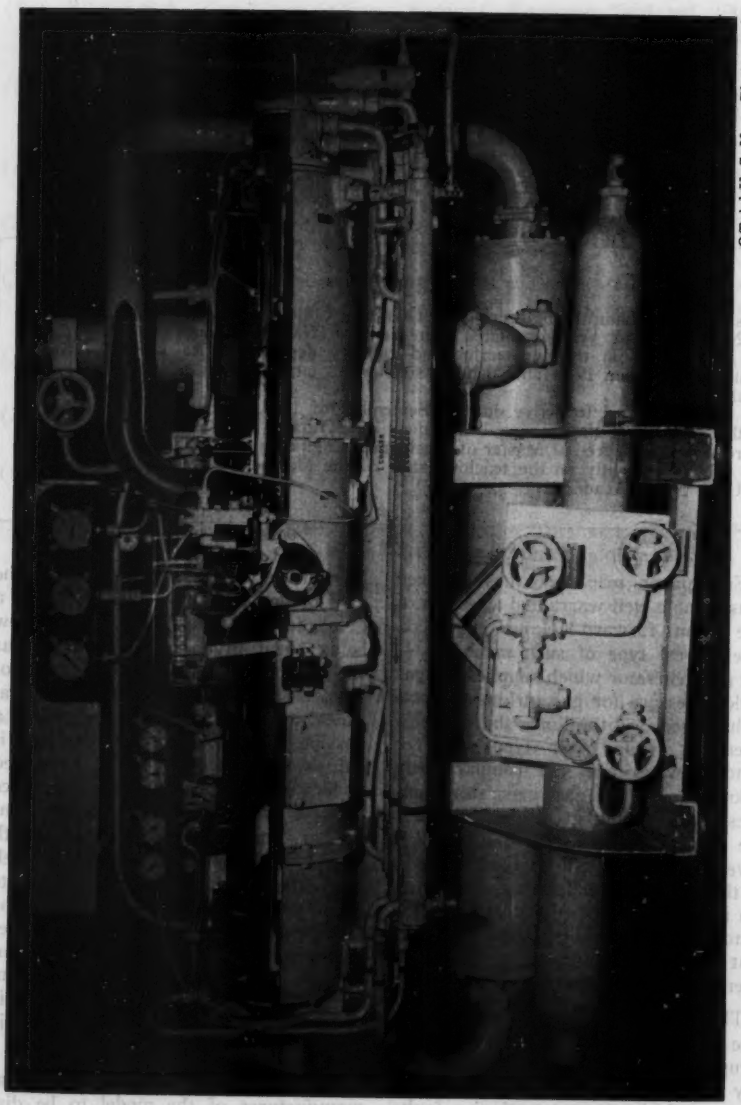
Free piston machinery has received considerable well warranted attention of late from forward-looking engineers. The newest type of such machinery—the gas generator which supplies hot gas under pressure for gas turbines—seems to have received the lion's share of this attention, while other types of free piston devices are still comparatively unknown. While gas generators unquestionably show great promise, they are still in the experimental stage of development and will probably remain in that stage for at least several years. On the other hand, little is heard of free piston air compressors, despite the fact that they have been proved by more than twenty years of actual use.

The disparity of interest between the free piston gas generator and the air compressor can be attributed neither to any marked miracles performed by the generator nor to any particular faults of the air compressor. The fact is that each is very good in its own class, and therein lies the crux of the matter. The field of gas turbines is one in which a great deal of time and money is now being spent in an effort to make gas turbines sufficiently practical and economical to permit their more general use. Since the gas generator offers a possible means of attaining this end, it enjoys the benefit of being presented under favorable circumstances and hence receives wide acclaim. The air compressor, however, has to compete with conventional compressors which are well entrenched and are performing quite satisfactorily. The mere fact that the free piston compressor offers improvements over the conventional types is not enough to attract widespread attention. It is therefore the main interest of this article to present the free piston air compressor.

Junkers, in Germany, was the original manufacturer of the model to be discussed here. This compressor, delivering air at 3,000 psi., was in use in German



Official U. S. Navy Photograph

FIGURE 1.—Sectioned Junkers Free Piston Air Compressor on Display in Model Room, Melville Hall, U. S. Naval Academy.

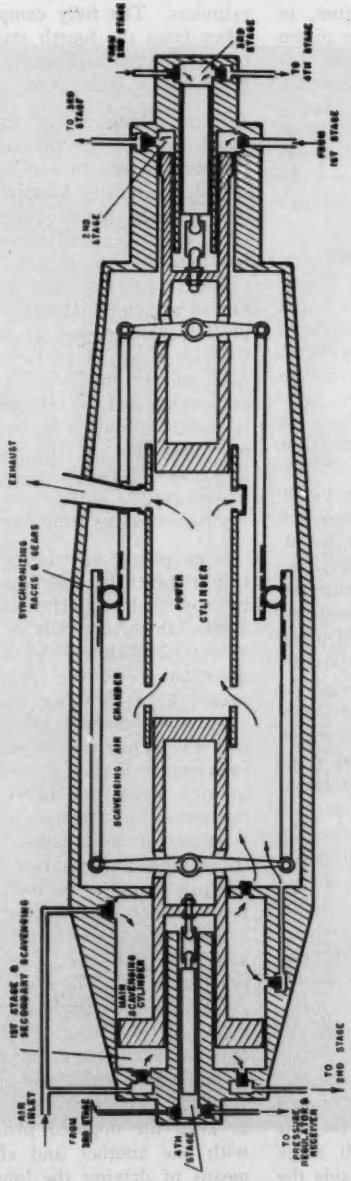
submarines during World War II. Since the end of the war, Worthington, in this country, has produced a free piston compressor with only very minor differences from the Junkers model, which is shown in Figure 1 and schematically in Figure 2. As the most direct method of explaining the principles of its operation, it is proposed that Figure 2 be used to trace the flow of air through the unit.

After passing through a strainer, the air enters the intake at the left and is supplied to both sides of a double acting piston which provides both scavenging air and the first stage of compression. On the instroke of this piston, its inboard side forces air under low pressure (normally about 2 psig.) into the annular scavenging air chamber in the central portion of the machine. Meanwhile, the outboard side of the piston is taking on air from atmosphere. When the piston moves outward again, its inboard side is on its suction stroke, taking air from atmosphere. The chamber on the outboard side of the piston has two sets of discharge valves. During the initial part of the outward stroke, scavenging air is discharged through the valves in the cylinder wall. At somewhat less than its half-stroke, the piston covers these valves and the first stage of compression begins, with the remaining entrapped air further compressed and discharged through the valves in the cylinder head. The air then passes through an intercooler to the second stage. The second stage cylinder is the outer and larger of the two concentric cylinders as shown at the right hand end of the unit. Air from the second stage goes through a two-pass intercooler and returns to the inner of the concentric cylinders previously mentioned, which is the third compressor stage. From here, the air passes through another cooler and returns to the left end of the unit to the fourth stage which is concentric with and inside the first stage and main scavenging air cylinders. The fully compressed air is taken from the fourth stage to the receiver through a pressure regulator (to be discussed later) and an aftercooler.

Backtracking a bit, the 2 psi. air which we left in the scavenging air chamber is used to scavenge the power cylinder centrally located between the two opposed power pistons which in turn are integrally connected to the four compressor pistons. The power cylinder operates on the two-stroke Diesel principle. It has a fuel injector in the cylinder wall at its center and there is a set of ports in the wall at each end of the cylinder, one set for scavenging and the other for exhaust. It is therefore similar in both appearance and operation to the opposed piston Fairbanks-Morse diesel engine cylinder. Scavenging is uniflow, and the exhaust is discharged to atmosphere.

Free piston devices get their name from the fact that they are not hampered by any such appurtenances as crankshafts, connecting rods, or flywheels. In an opposed piston model such as is being presented here, the pistons are connected to each other, but to nothing else. About midway between the power and compressor ends of each piston, still referring to Figure 2, there is a portion of each piston that never enters either the power or compressor cylinder and is thus always exposed within the scavenging air chamber regardless of the piston stroke position. In this exposed section, each piston is fitted with a yoke. To the ends of the yokes are attached gear racks, of which there are four in all. The racks of one piston are paired off with those of the other by means of two pinion gears diametrically opposite each other outside the power cylinder and within the scavenging air chamber. The racks and pinions serve to keep the opposed pistons in phase with one another and also provide a means of driving the lubricating, fuel,

FIGURE 1.—Sectioned Junkers Free Piston Air Compressor on Display in Model Room, Machine Hall, U. S. Naval Academy.



WORTHINGTON - JUNKERS
3,000 PSI
FREE PISTON AIR COMPRESSOR

FIGURE 2.

and cooling water pumps, and of actuating and timing the fuel injector.

When properly designed, the free piston air compressor has practically no vibration since the dynamic and pressure forces acting on one piston are exactly canceled by equal and opposite forces on the other piston. However, this is an ideal situation which is not completely achieved under all conditions of operation, such as the gradually increasing discharge pressure obtained as the air charge is carried to completion. There is therefore usually a small unbalance of forces which is equalized between the pistons through the racks and pinions. The unbalance is small enough however to permit making the synchronizing mechanism light and comparatively frictionless.

Having examined what happens when the pistons move back and forth, let us now see why they do so. Considering the cycle to begin with the pistons in their inner dead center positions, they are forced outward by combustion in the power cylinder. The expansion of the combustion products provides the energy necessary to overcome friction, to compress the air in each of the four compressor stages as the pistons move outward, and to store up kinetic energy in the pistons as their outward velocity increases. When the exhaust ports are uncovered, at about 75% stroke, most of the residual energy in the power cylinder is lost to the exhaust, but the pistons continue outward until their kinetic energy is completely absorbed by the work still being done in the compressor cylinders and the pistons stop at outer dead center. At this point, the compressor stages cease to discharge air, and the air in their clearance volumes begins to expand, forcing the pistons back inward. This inward motion of the pistons compresses the new charge of air in the power cylinder and permits each of the compressor stages to take on a new supply of air from the preceding stage or from atmosphere. Fuel is injected into the power cylinder as the pistons approach inner dead center and the cycle is then repeated. Incidentally, it is impossible for the pistons to strike each other at inner dead center because of the positive cushion supplied by the air in the power cylinder.

A pressure regulator is installed between the fourth stage aftercooler and the receiver to keep the fourth stage discharge pressure at 1600 psi. or greater regardless of the receiver pressure. When the receiver pressure is low, air is throttled in passing the regulator, whereas for higher receiver pressures, the regulator permits an unrestricted flow of compressed air. The regulator was found to be necessary because the cyclic frequency decreases with low fourth stage discharge pressures. With very low discharge pressures, the unit finally stalls due to insufficient energy in the compressor clearance air to return the pistons near enough to their designed inner dead center positions. This causes the compressor to stall either through improper functioning of the fuel injector or because of a misfire due to low compression temperature in the power cylinder. A fourth stage pressure of about 1500 psi is considered to be the minimum for satisfactory operation of the unit, with the 1600 pound setting of the regulator offering 100 pounds of extra assurance against "short-stroking."

As has been mentioned, there are very good conventional air compressors available at the present time. Most of the users of these compressors are perfectly satisfied with their performance, and since it is only human to be conservative where expenditures are concerned, the rush to replace existing compressors with free piston models will be anything but a mad one. This inertia will be a tremendous handicap for free piston compressors and it will probably be quite a few years before they come into

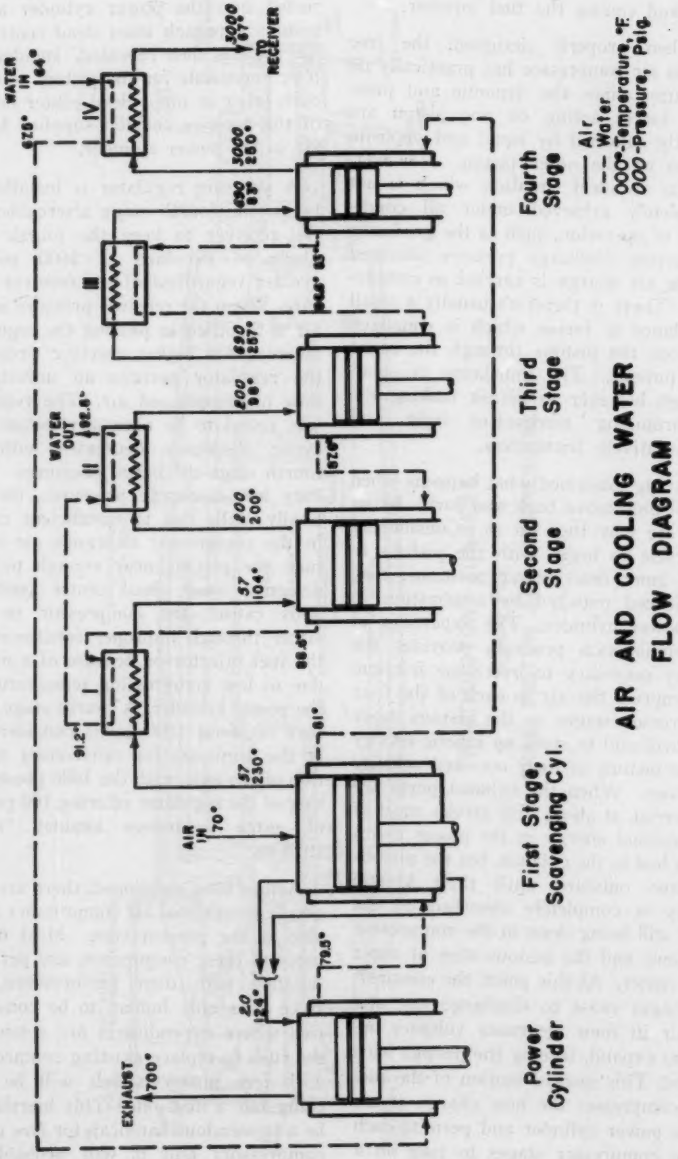

WATER IN
64°
IV
3000
87°
RECEIVER
3000
250°
WATER OUT
92.6°
II
III
I
AIR IN
70°
EXHAUST
700°
Power Cylinder
First Stage, Scavenging Cyl.
Second Stage
Third Stage
Fourth Stage
AIR AND COOLING WATER FLOW DIAGRAM
Air
Water
000-Temperature, °F
000-Pressure, Psig

FIGURE 3.

anything approaching genera. use. In applications where only a small volume of comparatively low pressure air is required, or where for one reason or another diesel engines cannot be used, the conventional compressors will probably continue in use indefinitely, but for most other applications, the free piston type should be seriously considered.

For general use, some of the more obvious advantages of the free piston compressor are:

(1) The prime mover and compressor are integral, making for simplicity, a small number of moving parts, minimum vibration, low maintenance, and reduced initial cost when produced in volume.

(2) Since there are no crankshafts or connecting rods, side thrust is eliminated, giving long ring, piston, and cylinder life.

(3) Little or no foundation is necessary. The unit can be moved from place to place or mounted overhead or on walls strong enough only to support its weight, thus conserving floor space.

(4) Size and weight are greatly reduced.

(5) It is superior to electrically driven compressors since it is unaffected by electrical power failure or voltage fluctuation.

(6) Its efficiency is high because of its small number of moving parts and lack of bearings and because it eliminates such "middle men" as generators, motors, and batteries.

These advantages are general. For specific installations additional advantages might be cited. A description of the operating data of the Worthington-Junkers compressor will give an indication of its performance relative to conventional types and may serve to support some of the advantages claimed for it.

(1) It requires about 8 by 3 by 3 feet and weighs approximately 1200 pounds.

(2) Rated capacity is 20 cubic feet per hour of air at 3,000 psig. and 80° F.

(3) Fuel consumption at rated conditions—17 pounds of diesel fuel per hour.

(4) Normal exhaust temperature—670 to 700° F.

(5) Frequency—750 to 950 cycles per minute, increasing with discharge pressure.

(6) Stroke length varies somewhat with operating conditions but averages about 8.5 inches for each piston.

(7) For each pound of compressed air delivered, 3.6 pounds of scavenging air are used, of which about 1.5 pounds are actually involved during combustion, and the remainder cools the exhaust and insures complete purging of the power cylinder.

(8) The compressor uses about 0.7 pint per hour of lube oil for lubrication of ten points: each of the six pistons, each of two synchronizing gears, and the actuating mechanisms for fuel oil pump and the lubricator.

(9) Five gallons per minute of cooling water are circulated. Figure 3 shows the flow of both air and cooling water for the whole unit and gives typical temperatures for the water and both pressures and temperatures of the air.

(10) About 25% of the energy supplied in the fuel is actually realized as useful work in the delivery of compressed air, making this a very efficient machine indeed. For an electrically driven compressor to be compared on the same basis, one would have to consider the losses in say a boiler, turbine, generator, motor, and finally the compressor itself, by which time there would be very little of the original energy left. While it is recognized that the efficiency of a unit requiring only a small percentage of the total power used in a large plant is of relatively minor importance, nevertheless, it is still an item to be reckoned with.

Probably the most important item in the description of the operation and performance of the free piston air compressor—the method of starting the unit —still remains to be covered. Since this compressor differs so greatly from most other forms of machinery, it has a starting procedure quite unlike that of any other piece of equipment. It is started with compressed air, with a 2.5 cubic foot flask at 1,000 psi. providing enough air for about eight starts without recharging. The flask retains its effectiveness until its pressure drops to about 350 psi.

The first step in starting is to move the pistons outward to a position near outer dead center. This is done by attaching a wrench to the shaft of one of the synchronizing pinions. The pistons are then locked in the outer position by a linkage in much the same manner as the sear holds the firing pin of a rifle. Figure 4 shows how the "sear" holds a lug which is an extension of the wrist pin of the synchronizing yoke at the scavenging end of the unit. After the pistons are locked, the wrench is removed.

A two position starting valve supplies all compressor cylinders with starting air at reduced pressure. In starting, the valve is manually placed in position 1 which supplies the second compressor stage with air at slightly less than 400 psi. This air flows through the intercoolers to the third and fourth stages and as far as the previously mentioned regulator in the fourth stage discharge line. The starting valve is then placed in position 2 which begins to supply the first stage with air. First stage pressure begins to build up until it reaches about 60 psi. at which time the piston locking mechanism is automatically released by the plunger shown in Figure 4. Because of expansion of air in the compressor stages, the pistons fly inward and the unit "takes off." It might be noted that on starting, the pressures in the compressor cylinders are not the designed operating pressures and therefore there is some end thrust—a maximum of about 1450 pounds—on the first stage of the pistons. However, after the small fraction of a second required for the first few strokes, all stages are operating at their normal pressures and from then on, end thrust and vibration are practically eliminated.

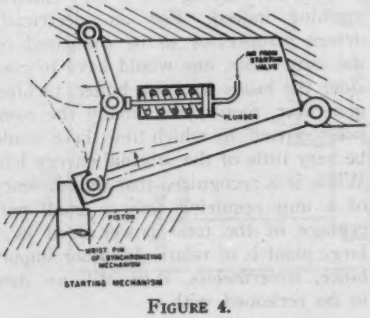


FIGURE 4.

The unit can be stopped merely by shutting off its fuel supply either manually, or automatically when the air charge is completed or when either pressure or temperature at any point becomes excessive.

Cycle frequency depends almost entirely on the fourth stage discharge pressure, increasing about one percent for each 100 psi. increase in discharge pressure. In addition to frequency, the other operating variable which determines capacity is the stroke length. Just as the discharge pressure was the major factor in determining frequency, so the amount of fuel injected per cycle has been found to be the only factor significance in determining what the stroke length will be. Thus, it may be seen that, for any discharge pressure above that for which the regulator is set, the free piston compressor runs at essentially constant frequency, with the air rate controlled by changing the throttle setting, which in turn controls the

stroke length and hence the amount of air taken on and discharged per stroke. This is a convenient situation because, even if there were a choice, it would be much easier to control delivery rate through stroke length rather than through frequency, since changes of stroke have by far the greater effect on the volume of air delivered. A 10% increase in frequency would give at the maximum a 10% increase in air rate, with the actual increase somewhat less than that amount because of increased air velocities, higher valve drops, etc. On the other hand, an increase in stroke length of ten-percent (assuming sufficient clearance in the cylinders to permit the increase) would raise the air rate by much more than ten percent because of the greatly increased volumetric efficiency obtained in the compressor cylinders with a slightly longer stroke.

As was mentioned in the opening paragraphs of this article, the original Junkers compressor was in use in German submarines during World War II. Germany has a brilliant history, at least as seen from the scientist's viewpoint, and her technical contributions to marine, naval, and in particular, submarine development have been outstanding. It therefore greatly behooves us to examine the particular advantages which motivated her engineers to use the free piston compressor in marine and, more especially, submarine installations.

The first and perhaps most obvious advantage is its reduction in size, since it occupies about 50% of the space required by a comparably rated conventional compressor with its electric motor. No one need argue the merits of anything which will make additional space available in a submarine, and when any such extra space is obtained, the submariner has learned to make better use of it than the average person could get from three times the space.

Since the F.P. compressor has its own prime mover, it is not dependent on the submarine's electrical system. This is advantageous in several respects. The auxiliaryman who operates the compressor is able to do all required maintenance and repair work on the whole unit without the assistance of an electrician's mate. One source of electrical ground is eliminated. Probably the most important aspect of elimination of electric drive is that a battery charge and an air charge can be conducted independently of each other. With motor driven compressors, the high voltage obtained during the latter stages of a battery charge would be likely to burn out the compressor motors if an air charge were also in progress.

Ordinarily, the increased efficiency of an F.P. compressor would be a minor consideration, but in a submarine, it assumes additional importance. During wartime, anything that permits a submarine to remain on station even an extra day increases its combat effectiveness. The free piston compressor might very well do this through the saving in fuel it permits. It would also conserve some of the energy of the batteries for the more important function of providing submerged propulsion.

Prior to the German use of the snorkel, the question would not have arisen, but now it is important to know how the free piston compressor performs under snorkeling conditions. In a nutshell, it can be said that the operation of the F.P. compressor is hampered by snorkeling, but not as badly as that of a conventional compressor. Lest this be considered as only a half-hearted endorsement, let us examine the operation of both the conventional and the F.P. compressor under snorkeling conditions.

As far as the conventional compressor is concerned, the only difference it

notices while in snorkel operation is the slight vacuum in the boat. This vacuum reduces the compressor capacity mainly by the reduced density of the inlet air, and secondly because of reduced compressor volumetric efficiency, since it now operates over a greater pressure differential. Since the first of these factors, reduced air density, will by itself cause a capacity reduction in direct proportion to the vacuum in the boat, the conventional compressor loses capacity at a rate greater than the reduction of absolute pressure within the submarine while snorkeling. It could therefore probably be relied upon for no more than 75 to 80% of its normal capacity when operating under the largest pressure reduction likely to be obtained under extreme snorkeling conditions.

The F.P. compressor must contend with this same pressure reduction at its inlet, and at the same time, it has the additional handicap of having to discharge its power cylinder exhaust against a higher back pressure than would be the case with operation on the surface.

So, at first glance, it might be expected that the F.P. compressor would suffer an even greater capacity reduction while snorkeling than would the conventional type. But obviously such must not be the case, or the author would never have so eagerly brought up the subject. Figure 5 shows an indicator diagram for the first stage of compression in both normal and snorkel operation. To help bring out the difference between the two types of operation, the diagram has not been drawn to exact scale. First stage discharge pressure has been assumed to be the same for both cases. The "step" in the compression line is caused by the discharge of air into the scavenging air chamber during the first part of the compression stroke. When snorkeling, the increased exhaust back pressure requires a corresponding increase in scavenging pressure. This, coupled with the reduced air inlet pressure, means that there will be a considerably smaller weight of air supplied to scavenge the engine from this cylinder. However, since the output of the main scavenging cylinder is not so adversely affected and since it is still supplying its total output to the power cylinder, there is plenty of scavenging air available for satisfactory power cylinder operation.

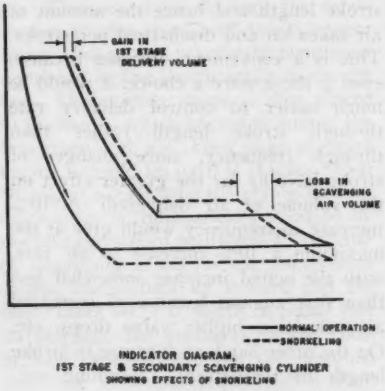


FIGURE 5.

Now on Figure 5, note that at the time the scavenging valve is covered, the pressure in the first stage is greater than it is with normal operation. This means that the F.P. compressor discharges a larger and larger amount of compressed air per cycle as snorkeling conditions become progressively more extreme, which is the opposite to the effect obtained in the conventional compressor. About the only drawback of the free piston compressor in snorkeling is the reduced scavenging air. There is still an adequate amount of air available for combustion, but the exhaust temperature is considerably increased. To keep the exhaust temperature from becoming excessive, the flow rate of the cooling water can be increased, or the amount of fuel injected can be reduced.

Another means would be to bleed additional scavenging air from the discharge of one of the compressor stages. The latter two steps would be less desirable than the increase of cooling water flow, since they would both have the effect of reducing the output of the compressor.

On the other side of the ledger, one slight advantage of the conventional compressor in a submarine is that it can be used to help revitalize the air in the boat during long periods of complete submergence. Fresh air is bled into the boat from the air banks while the stale air is returned to the banks through the compressor. There are also times when it is helpful to remove built up air pressure from the boat by means of the air compressor. Minor though these functions may be, the fact remains that they could not be performed by an F.P. unit due to the necessity for eliminating its diesel exhaust. Possibly these functions could be dispensed with; if not, some other means than the free piston compressor will have to be devised to accomplish them.

The main purpose of this article—the introduction of the free piston compressor—has now been fulfilled, but the possibilities for the future of other very similar free piston devices are also worthy of mention. With comparatively minor changes, the compressor described could be made to compress other gases, such as refrigerants. Another variation might be used in a pumping station in a natural gas pipeline. In this case, it could undoubtedly be designed to use as a fuel some of the gas it is pumping. The compression of any other fluid except liquids might make equally appropriate use of a free piston compressor.

The author feels that the inauspicious history of the free piston compressor during the past twenty years has been due almost entirely to a general lack of knowledge not only of its potentialities but even of its very existence. The radical differences of a new type of machinery by themselves should certainly prove no handicap in this rapidly moving era. So perhaps within the next few years the free piston air compressor will be accorded its rightful place along with other technological developments even more radical in nature. It is only through acceptance of such changes that progress is made.

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ON THE CALCULATION OF THE THERMODYNAMIC ICE POINT

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SUMMARY

The computation of the ice point on the Kelvin Temperature scale is surveyed. The standard method of solving the equation of the constant pressure gas thermometer is reviewed.

Consideration is given to an alternate procedure in which $\frac{\mu C_p}{T^2} = f(T)$ is formulated, facilitating the integration of that term. Certain advantages accrue to this method, principally the fact that the work of different observers may be compared on a more basic level.

The ice point is computed for air at one atmosphere.

I—INTRODUCTION

The determination of the thermodynamic ice point has held a classic position in the field of thermometry. The problem of reducing gas thermometer readings to the Kelvin scale has been widely explored in the literature, and at least ninety-six determinations of the ice point have been reported. This paper is concerned with a method for computing the thermodynamic ice point from data on the constant pressure gas thermometer.

II—THEORY

The specific heat at constant pressure and the Joule-Thomson coefficient are related through the well known equation:

$$\mu = -\frac{1}{C_p}\left(\frac{\partial h}{\partial P}\right)_T \qquad \ldots\ldots(1)$$

Also, from the equations of thermodynamics:

$$\mu = \frac{1}{C_p}\left[\, T\left(\frac{\partial V}{\partial T}\right)_p - V \,\right] \qquad \ldots\ldots(2a)$$

$$\mu C_p = T\left(\frac{\partial V}{\partial T}\right)_p - V \qquad \ldots\ldots(2b)$$

but:

$$T\left(\frac{\partial V}{\partial T}\right)_p - V = T^2\left[\frac{\partial}{\partial T}\left(\frac{V}{T}\right)_p\right]$$

hence,

$$\frac{\mu C_p}{T^2} = \frac{\partial}{\partial T}\left(\frac{V}{T}\right)_p \qquad \ldots\ldots(3)$$

Integrating between the ice point and the steam point:

$$\frac{V_s}{T_s} - \frac{V_i}{T_i} = \int_{T_i}^{T_s}\left(\frac{\mu C_p}{T^2}\right) dT \qquad \ldots\ldots(4)$$

Noting that $T_s = (100 + T_i)$, and rewriting:

$$T_i = \frac{100\, V_i}{(V_s - V_i)} + \frac{(100\, V_i)\, T_i (T_i + 100)}{(V_s - V_i)(100\, V_i)} \int_0^{100} \frac{\mu C_p}{(t + T_i)^2}\, dt \qquad \ldots(5)$$

In practice, v_s and v_i are carefully determined, and values of μ and C_p are measured in the temperature range between 0°C. and 100°C. For chosen values of T_i, the function $\mu C_p/(t + T_i)^2$ is plotted against t, and the integral is solved graphically.

Zemansky[1] points out that at one atmosphere the term $100\, v_i/(v_s - v_i)$ is very nearly equal to the Kelvin ice point, and that the second term constitutes a correction amounting to less than one degree. Three significant figures in the correction term are sufficient for most purposes, as an error in the third figure of the correction term becomes an error in the sixth figure of the ice point temperature. Hence the factor outside the integral may be expressed as $(273)^2 \times (373)/100\, v_i$, and the solution for T_i simplified to some extent.

The procedures outlined above have sprung from practical considerations. Usually the Joule-Thomson data and the specific heat data are available as instantane-

ous values at specified temperatures. The individual worker has used his own data for μ and C_p, prepared the graphical integration, and reported his own value of the probable ice point temperature.

Attempts have been made to express the effects of pressure and temperature on the Joule-Thomson coefficient.[2, 3] These relations are complicated, and, except for very narrow limits, are not dependable. However, in the isobaric case, μ may be represented as a simple function in rising powers of T. The specific heat at constant pressure may be expressed satisfactorily as a power series in T for the temperature range under consideration.

Because μ and C_p usually contain at least three terms each, the prospect for algebraic formulation of μC_p appears discouraging. A more logical solution is to be found by plotting the instantaneous values of the product μC_p against temperature, then determining the equation of the resulting curve. This procedure was tried for three of the gases normally used in thermometers: nitrogen, air and helium. For comparison purposes two other gases, argon and hydrogen, were also studied. In order to plot the data on a single graph the reduced function

$$\varphi = \left[\frac{(\mu C_p)_T}{(\mu C_p)_0}\right]$$

was used. The plot of φ against t is shown in Fig. 1.

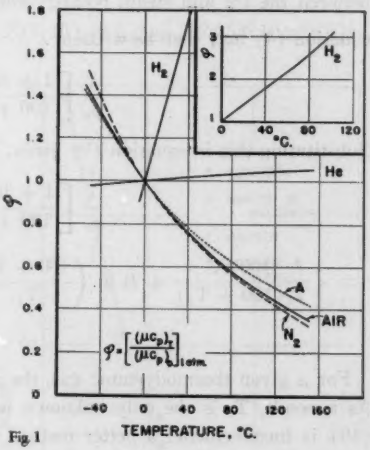

Fig. 1

It is interesting to note that the curves are smooth and continuous, suggesting that μC_p and T are related by equations which are not especially complicated. In general, the relationship may be expressed with sufficient accuracy by a power series in T. From equation (1), the product μC_p may be written as:

$$-\left(\frac{\partial h}{\partial P}\right)_T = A + B(T) + C(T)^2 + D(T)^3 \quad \ldots\ldots\ldots(6)$$

The function required for the integrand is then:

$$\frac{\mu C_p}{T^2} = \frac{A}{T^2} + \frac{B}{T} + C + D(T) \quad \ldots\ldots\ldots\ldots(7)$$

Integration yields:

$$\int_{T_1}^{T_2} \frac{\mu C_p}{T^2}\, dT = \left[\frac{A(T_2 - T_1)}{T_2 T_1} + B \ln. \left(\frac{T_2}{T_1}\right) + C(T_2 - T_1) + \frac{D}{2}(T_2^2 - T_1^2)\right] \quad \ldots\ldots\ldots\ldots(8)$$

or

$$\int_{T_i}^{T_s} \frac{\mu C_p}{T^2}\, dT = \left\{ \frac{100\,A}{T_i(100 + T_i)} + B \ln. \left(\frac{100 + T_i}{T_i} \right) + 100\,C + \frac{D}{2} \left[10{,}000 + 200\,T_i \right] \right\} \quad \ldots\ldots\ldots\ldots (9)$$

The volume occupied by the gas when the thermometer is at the steam point is given by: $v_s = v_i\ (1 + 100\ \alpha)$, where α is the mean coefficient of thermal expansion between the ice and steam points. Noting that $v_i = \frac{1}{\rho_o}$, the left hand member of equation (4) may then be written:

$$\frac{1}{\rho_o} \left[\frac{1 + 100\,\alpha}{100 + T_i} - \frac{1}{T_i} \right]$$

Substituting this in equation (9) gives:

$$\frac{1}{\rho_o} \left[\frac{1 + 100\,\alpha}{100 + T_i} - \frac{1}{T_i} \right]$$

$$= \left\{ \frac{100\,A}{T_i(100 + T_i)} + B \ln. \left(\frac{100 + T_i}{T_i} \right) + 100\,C + \frac{D}{2} \left[10{,}000 + 200\,T_i \right] \right\} \quad \ldots\ldots\ldots\ldots (10)$$

For a given thermodynamic gas, the constants A, B, C, D, α, and ρ_o are known. As a result, T_i is the only unknown in equation (10). The algebraic solution of (10) is inconvenient; a better method would be to transpose the equation setting it equal to zero, assume values of T_i, and plot the residues. The ice point temperature is then taken as that for which the residue is zero.

III—CONCLUSIONS

(1) As the product μC_p is simply related to temperature between the ice and steam points for the media used in gas thermometers, it is possible to formulate $\frac{\mu C_p}{T^2} = f(T)$. Integration of this term is not difficult; the resulting equation being unique in T_i.

(2) The advantage of this method lies in the fact that μC_p data may be evaluated when $\mu C_p = f(T)$ is formulated. The work of many observers may be assessed at this point, and it is probable that such comparisons of μC_p data could lead to a value of the ice point temperature which would be more acceptable than that obtained by averaging the ice point values of the separate workers.

IV—APPLICATION

To illustrate the technique described in the theoretical discussion, the ice point has been computed from data on the constant pressure air thermometer. Joule-Thomson data were obtained from the work of Anderson,[4] Hausen,[5] and Roebuck-Murrell.[6] As a check, the Beattie-Bridgeman[7] equation was differentiated and substituted in equation (2b). The three experimental sources of μ data were multiplied by C_p values taken from the latest Bureau of Standards tables.[8]

The instantaneous values of μC_p were plotted against temperature as shown in Fig. 2. The data were found to satisfy the equation:

$$\mu C_p = 22.5467 - 147.432(10)^{-3}(T) + 357.733(10)^{-6}(T)^2 - 307.553(10)^{-9}(T)^3$$

This expression was divided by $(T)^2$ and integrated to give the left hand side of equation (10). Solving by the residue method gave an ice point temperature of 273.144° Kelvin. Although this value is lower than the currently accepted ice point (273.165° K) it is within the range of some of the recent experimental data. In the table below, some of the recent ice point values are compared:[6]

Observer	*Date*	*Ice Point*
Keesom	1927	273.118
Heuse	1926	273.130
Keesom	1933	273.147
Heuse	1929	273.138
Kinoshita	1937	273.188

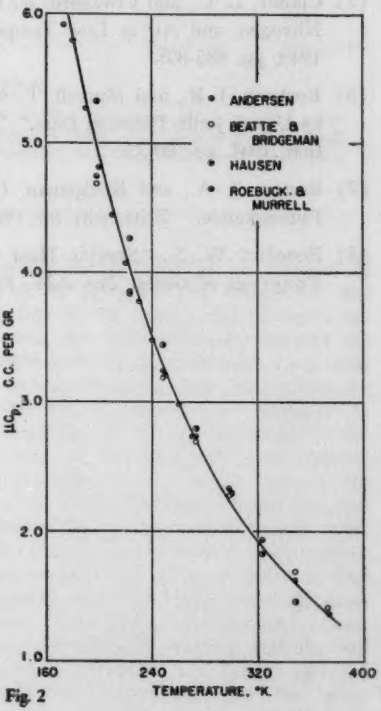

Fig. 2

V—ACKNOWLEDGMENT

The author wishes to thank Professor J. R. Roebuck, of the University of Wisconsin, and Professor T. A. Murrell, of the University of Illinois, each of whom offered criticism and suggestions which were embodied in the final draft of the paper.

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COMMENT ON MARINE DIESEL PLANT FOR RESIDUAL FUEL PERFORMANCE

LAURIN EMANUELE

THE AUTHOR

was born in Genoa in 1913. He has been in the Merchant Navy service since 1930. Graduated Naval Engineer at Trieste in 1944. At present time Second Engineer on M/V *Sakura*, % Messrs. NAESS, MEJLAENDER & CO., Inc., New York, N. Y.

The choice of propelling machinery for the replacement of the ineffective mercantile tonnage, prevalent during this peacetime, refer in the most part to the oil engine and the geared steam turbine. Observing the world naval output it is not difficult to foresee that in the near future the motor ship will reach the position which was for so many years previously held by the reciprocating steam plants.

The valuation of the technical odds concerned in this or that engine is not to be made here for discussion purpose. Furthermore, it is known that the best thermal efficiency of the motor vessel is *commercially* reduced in relation to current boiler fuel and diesel oil price. If in the past, diesel oil was almost universally adopted, today marine diesel plants capable of replacing steam plants with the same bunker fuel are desirable, and world-wide technical publications advise the shipowner to operate their motor vessels on boiler oil to effect a saving in the fuel bill of between 20 and 35 percent. The reason for this policy is the commercial situation, because the ever-increasing demand for light fuel for land purposes, not only increases the price, but will confront the Oil Administration with difficulty in supplying the market, especially if, because of the replacement of obsolete steamships by new motor vessels, the actual residual demand should fall. Engineering progress has apparently failed to foresee this situation since modern oil engines are more sensitive to fuel quality than those built some 25 years ago. Nevertheless several motor vessels are satisfactorily running with the fuel sold as Bunker "C" Grade and the technical departments of the shipowners concerned have already published their experimental data and the improvements devised to attain this aim with the existent vessels. For new construction, as far as I am aware, a design of motor vessel exclusively specified to burn residual fuel is still desirable.

To discuss the main problems that such a design must overcome it is useful to confront the basic requirement for acceptable fuel for diesel use (condensed in the following six perquisites) with the deficiencies of a lower grade bunker:

(a) Diesel fuel must be capable of being pumped at winter temperature without preheating.

(b) Must not contain water or sediments.

(c) Must not cause deposit on injector nor in combustion chamber.

(d) Must not contain corrosive impurities, nor produce them as products of combustion.

(e) Must not cause sludging of lubricating oil.

(f) Must ensure easy starting under winter weather conditions.*

(A) When the fuel shipped is of high viscosity, steam coils in the fuel tank are needed. Since it is the modern usage to store the oil in the hull's double bottoms, heating is not easily attained by the modest steam generators usually fitted on motor vessels, owing to dispersion though the bottom which is surrounded by a wide wall between fluids having different temperature. To minimize such dispersion of heat, the double bottom tanks of a motor vessel built to run on boiler oil should be designed like the "compensation tanks" of some war vessels, where the suction is placed in the top, the level being maintained constant by replacing the consumption with sea water. This allows heating only of the upper share of the tank with less dispersion on account of the insulating effect of the lower liquid strata. Besides this, the natural possibility of containing fuel or water ballast as well in the same double bottom, is profitable for the autonomy and the stability of the ship, avoiding the troubles experienced with common plants when nautical exigencies require the filling of empty oil reservoirs with water.

(B) The residual fuel is specified with a sediment tolerance not acceptable for Diesel use. The purification technique is well improved in the marine industry, but for boiler oil treatment the solid impurities separated may be of such a degree to make this process difficult by self-cleaning separators, especially when in rough seas the oil shakes up the settled sludge into the supplying double bottom. In the plant proposed in (A) above, the sludge deposited in the tanks cannot reach the suction pipe in heavy seas. Even so, settling tanks, with steam coils, are always essential to relieve the purifiers. The method of purification by washing with a proper dosage of warm fresh water, mixed with the oil when being delivered to the separator, in order to remove every soluble impurity (e.g., sodium chloride) is to be recommended also if a further drying up is necessary.

(C) The common means to have effective boiler oil combustion in Diesel cylinder is to raise the temperature as near the flash point in relation to viscosity, by a steam heater fitted on the inlet pipe near the fuel pump; but the solution should be such that the oil has the required physical properties when it enters the atomizer body as occurs in the Cummins injection system. The application of this technique in propelling engines will avoid the trouble of oil temperature failure that occurs in the piping from fuel pumps to fuel valves, thus heating will be needed only as a standby and only at lower levels at full power.

The fuel heating equipment for diesel must be a very suitable one regarding

* Fuel for Navy's Diesels, by Rear Admiral H. G. Bowen. JOURNAL A.S.N.E., vol. 49, Aug. '37, page 452.

safety and must ensure thermostatic control of delivery however the consumption varies. A steam heater will scarcely fulfill the thermostatic condition while electric resistor heater increases the risk of oil cracking; this leads to the proposal (to be confirmed by experiments) of an electric heater with the resistor elements surrounded by a watertight jacket of distilled water, with steam dome of wide surface, radiating the heat to the oil. Each variation of temperature of the fuel is proportional to the variation of pressure in the steam jacket, a pressure switch which controls the power to the resistor, by solenoid contractor, can be set to keep the temperature at required value. Where steam coils in the fuel daily service tank are fitted, the kW. spent for this purpose should not affect the generator's load.

(D) With reference to the damage of corrosive impurities of products of combustion in the 4-stroke engine, the chief attention is due to the exhaust valves which are more liable to experience the blow pipe effects; in 2-stroke engines an excessive cylinder liner wear is to be expected. To make this engine capable of operating on heavy fuel oil, the cylinder gears must be corrosion proof. Since chrome hardened cylinder liners have 12 years of background, it is desirable that this improvement extend to the 2-stroke engine designed for boiler oil.

(E) On residual fuel performance an increased amount of contamination of the crankcase oil due to the blow-by of combustion sludge is experienced. The engine with a partition between the cylinder in the crankcase with a stuffing box through which the piston rod passes, is insensitive to fuel quality, because no products of combustion can reach the lubricating oil.

(F) The method of starting and maneuvering the engine on light fuel and shifting to boiler oil at full power, requires a double fuel system and an increase in the construction bill. In the motor vessel where bunker "C" is shipped for every purpose no starting trouble has been recorded, the preheating of the circulating system of cylinder's jacket is recommended.

Every development in the performance of motor vessels on boiler fuel is worthy of attention. The sea trade economy is an important factor in raising the standard of living of mankind. Nevertheless the engineering staff still has some reasons for skepticism. This comment has been written with the hope that the discussion upon the innovations personally devised may be profitable and encourage a complete study by authors who are interested.

THE STATE OF THE NATION— IN SHIPBUILDING

ACKNOWLEDGMENT

Rear Admiral HOMER N. WALLIN, USN, Chief of the Bureau of Ships, Navy Department, and Coordinator of Shipbuilding, Conversion and Repair for the Department of Defense, delivered this speech at the banquet of the Spring Meeting of The Society of Naval Architects and Marine Engineers at New Orleans, Louisiana, on 2 May, 1952.

I am pleased indeed to meet with fellow members of the Society of Naval Architects and Marine Engineers in New Orleans, one of the great historical cities of the South, famed for its unique atmosphere and for its unusual accomplishments. The gulf states have contributed mightily to our national defense, and in World War II particularly, they participated in the fullest measure to America's marvelous record of shipbuilding.

Because of the selective membership of this Society, there exists among us a spirit of friendship and of pride, probably because under our national economy over the past 50 years at least, people who have devoted themselves to shipbuilding have eked out a rather precarious living. Yet shipbuilding and ship operation are a substantial part of the life blood of our country—not only as the key to our national defense in time of frequent need, but also in normal times when foreign trade constitutes the difference between prosperity and lack of it. Too bad, isn't it, that for some strange and unusual reason shipbuilding in our country is treated like a fire department, except that it isn't maintained well between fire alarms? If I may paraphrase Kipling:

> God and the Shipbuilder we adore
> In time of danger, not before;
> With the danger past, and all things
> righted,
> God is forgotten and the shipbuilder
> slighted!

All of us who participate in any manner in the construction of ships have a right to be proud of our association with the ancient and honorable craft of shipbuilding. Without ships and shipping no nation in history ever attained and retained greatness. The prosperity and well-being of nations and empires have always been intertwined with the accomplishments of shipbuilders and sailors.

Now as to ships, of all the complicated machines built by modern man, none can stir the emotions and the imagination as can the completed ship with her power, her grace of line, and

her air of far-off places and stirring events. Nearly 100 years ago the poet John Ruskin termed a ship of the line "the most honorable thing that man, as a gregarious animal, has ever produced." "Into that," said Ruskin, "he has put as much of his human patience, common sense, forethought, experimental philosophy, self-control, habits of order and obedience, careful patriotism, and calm expectation of the judgment of God, as can well be put into a space of 300 feet long by 80 broad. And I am thankful to have lived in an age when I could see this thing done." So said John Ruskin long before the newer inclusions of radio, radar, sonar, gadgets and gimmicks of every description and complexity. Truly a ship is a marvelous thing! I hope that all of you will seek the opportunity of visiting the great new passenger liner *United States.* She is by all odds the finest ship of her type yet built, and a great tribute to American ingenuity, as reflected by her designer and her builder.

Shipbuilding has become one of the most complex industries of a bewildering, industrial society. We are all professional shipbuilders or associated with shipbuilders, and we can profitably discuss our problems only if we consider the hard facts of today, rather than linger over the romantic episodes of the past.

Shipbuilding in the United States has been of late years a lagging and faltering industry, perhaps a fitting heir to an uneasy past. One hundred years ago, our industry was building the sharp-prowed, tall-masted clipper ships with which American operators dominated the commerce of the high seas.

American shipping went into a decline about 1855, when the clipper ships were at the height of their prosperity. The advent of the steamship is often cited as the cause, but the whole story forms a complex pattern. Other countries started to pour heavy subsidies into steamship lines. The War Between the States gave foreign operators a chance to get a strong grip on the best world trade routes. And when the war was over, America's interest turned westward to the challenge of building and developing a new continent. Capital as well as manpower was attracted to new forms of enterprise in the interior. This was the Indian-fighting, not the seafaring, era of American history.

The eventual result was that by 1914, foreign ships were carrying more than 90 percent of American foreign trade. And when Europe plunged into war that year, their ships were withdrawn from world trade, American goods piled on our wharves because the foreign ships we had depended upon no longer were interested in our business.

We started a hasty shipbuilding program that was not only expensive but was begun too late to be of any real help to us during the war. When war ended, national interest in American shipping almost disappeared and our yards languished once more. While the shipyards of the world built nearly 18,000,000 tons of shipping of all kinds from 1922 until 1932, the United States contributed only 5% of that total, and not a single ton did we build for dry-cargo foreign trade.

This paucity of performance was in spite of the fact that the principle that a merchant marine is necessary for an adequate national defense had been recognized and put on paper in the Merchant Marine Acts of 1920 and 1928. But the assistance provided through government construction loans and mail subsidies failed to foster a shipbuilding program. The Merchant Marine Act of 1936 replaced this form of assistance with direct help in the form of construction and operating subsidies. This act served to get construction started on an initial 50-ship program in 1937. Even so, the industry as a whole was still in the doldrums when this country started to rearm in 1939.

Yet the shipbuilding industry responded to the World War II crisis with the greatest accomplishments in maritime history. In 1938, the industry was creaking along with only a handful of ships on order. In three years from 1942, it built over 1000 naval ships and nearly 5000 merchant ships. No wonder shipbuilding has been called a feast or famine industry in our country!

It is sad to realize that from the heights of 1945 the industry was allowed to drop back to the famine stage again. Five ships a day were built at the peak of the wartime effort. A year later our shipyards were nearly idle. By 1948, less than three ships a month were being built in this country. The empty shipways were rotting away and the skilled shipbuilders had drifted off to other industries.

I wonder if we can point exclusively to sources outside the shipbuilding industry when we try to account for its frequent anemic condition. Perhaps part of the blame rests with the industry itself. Individually, American shipbuilders are energetic and know their business, but apparently many of them would rather hang separately than hang together. If it is cohesion that is lacking, would it not be a boon to a great nationwide industry if that industry could act and speak with a single voice on general shipbuilding matters as an industry essential to the country's welfare. I believe it is true that this great Society of Naval Architects and Marine Engineers has done more than any other organization to develop and maintain a spirit of pride in and devotion to the shipbuilding craftsmanship of America. But a technical organization is not enough.

At this point, I want to explain why I, a naval officer, am spending so much time in discussion of merchant shipbuilding and shipping. Basically, it is this: Not only is the Merchant Marine a veritable fourth arm of defense in its own right, but a healthy peacetime shipbuilding industry is indispensable to national defense. A nucleus of active, prosperous private shipyards is the base upon which the Navy must rely for emergency shipbuilding in case of mobilization. Without a healthy shipbuilding industry, America's mobilization reserve could be fatally deficient.

America leads the world in the production of steel and machinery and other industrial products. In merchant shipbuilding we rank fourth. We trail behind Great Britain, Sweden and Germany. Close behind us are Holland and Japan. Why do we have such a disparaging comparison? Principally because foreign governments have long recognized their merchant fleets as instruments of national policy and national prosperity, and have subsidized them heavily. The foreign ship operator has far more to gain and far less to lose than his American counterpart.

The result of the various and complex circumstances which seem to prevail has brought few rays of sunshine to American shipbuilders and operators. It is not a pleasing prospect for anyone trying to plan for America's long-range future on the seas. Here are some of the facts:

During 1951, only 38 percent of American export cargos and 40 percent of our imports were carried in American bottoms.

The only sea-going dry-cargo vessels now under construction in the U.S. are the 35 Mariner-class ships being built for the Maritime Administration. But that seems to be the end of this particular program inasmuch as the present Congress is not authorizing or appropriating for additional ships.

In foreign shipyards, over 800 modern dry-cargo vessels are now under construction or on order for foreign-flag shipping operators.

Also in foreign yards, 178 vessels, including 46 dry-cargo vessels, are now

being built for American or affiliated interests. All of these vessels will be registered under foreign flags when they are completed, and will be manned by crews of foreign nationality. Most of them will be registered in Panama and Liberia, but used in the foreign commerce of the United States.

Fortunately, the latter vessels are not lost to the United States for national defense purposes. They may be requisitioned by this country during an emergency if they are owned by United States citizens.

Throughout the world, 140 passenger or passenger-cargo vessels are under construction or on order for foreign flag shipping operators. In comparison, only one passenger liner, the *United States*, is under construction for an American shipping operator. Not a single ocean-going passenger vessel is on order in this country.

The general condition of the American passenger-carrying fleet is indicated by the fact that, in 1939, 131 passenger and combination passenger-cargo vessels were in operation under the American flag. They had a capacity of over 38,000 passengers. Today's passenger and combination fleet has only one-fourth this many vessels, with only 30 percent of its pre-war passenger capacity.

This state of affairs in the American merchant marine is directly reflected in the condition of our private shipbuilding industry, which has been limping at a fraction of its maximum capacity. At the beginning of the current calendar year, the United States shipbuilding and ship repair industry employed in private shipbuilding and boatbuilding and repair yards only about 111,000 production workers on the payroll. This total included those employed on the naval program, and the total is less than 20% of the capacity for normal operating conditions for efficient production.

It is taken for granted that commercial shipbuilding must be the backbone of America's shipbuilding industry, but private yards have also come to depend upon a certain amount of naval work. It is fitting that private firms should participate in naval work to the greatest degree possible. Moreover, it is necessary that the Navy assume some portion of the responsibility for maintaining the basic health of an industry upon which it relies so heavily in time of mobilization.

Naval shipbuilding programs since the war have been relatively modest. We have tried to distribute the work of these programs in a way that would be of most benefit to private shipbuilders. Probably one of the most frequently used terms in my vocabulary since becoming Co-ordinator of Shipbuilding for the Department of Defense, has been the expression, "broadening the base of shipbuilding procurement." We want to spread the work to as many operating shipyards as possible, in order that there shall be a firm base for rapid expansion if the need arises. This doesn't mean that I stand like an all-powerful Jupiter on his mountain-top, dispensing contracts where I think they will do the most good. The Bureau of Ships does its business with the taxpayer's dollar in mind. We are trying to buy quality ships at the lowest possible cost to the government. This has meant competition as the determining factor in awarding contracts. We do not wish to allocate ship contracts because allocation almost always introduces contractural problems and means higher prices, as we found out in World War II.

We also feel that the best kind of a contract is a fixed-price contract. We use a fixed contract whenever we have assurance that the price is right. We avoid where possible, the so-called cost-plus and flexible type contracts because we feel the fixed-price provides the best

incentive to a producer to get the job done on time. We are highly pleased with the results we have obtained through our procurement methods. We have obtained good prices. We have also realized what we feel is a reasonable geographical distribution of our work.

We are pleased that a considerable amount of the work will be done in the excellent facilities of the Gulf Coast region. It is estimated that less than ten percent of the U.S. shipbuilding capacity is located in this area. Yet, 40 of the 289 naval vessels contracted for since 1 July 1950 are being built by yards in this area.

One hundred seventy (170) of the 841 Army and Air Force vessels contracted for during this period are also being built by yards in this area.

The Ingalls Shipbuilding Company of nearby Pascagoula, Mississippi, is now working on four landing ships, dock, for the Navy. These are good-sized ships, with an over-all length of 510 feet, and of 6500 tons displacement. They are of slightly different design than the World War II LSD, and will take advantage of some of the lessons of that conflict. The Ingalls Company will handle the design work as well as the actual construction on these vessels. This yard is also building a newly designed icebreaker and several LST's.

Ten minesweepers, seven 100-foot tugs, 32 65-foot cargo-passenger boats, and many barges are included in the work underway at the Higgins Yard here in New Orleans, and Avondale Marine Ways at Westwego has contracts for construction of a number of barges, 29 tugboats and three patrol craft. Lighters and barges of various types are being built by the Hub City Iron Works in Lafayette and a Du Pont Yard at Morgan City. Along the nearby Texas coast, the Burton Construction and Shipbuilding Company of Port Arthur and a Du Pont Yard in Houston are also working on barges and lighters. At Port Houston, Texas, the Port Houston Iron Works is building eight LSU's and several lighters.

About $2,400,000,000 is being spent under the 1951 and 1952 shipbuilding and conversion programs. Seventy percent of this amount will be spent for ships to be built in private yards. Fifty-nine shipyards in all parts of the United States now have contracts for the construction and conversion of vessels for the Navy or foreign military aid. The number of shipyards rises to more than 100 when the small craft and small boats are included.

The Bureau of Ships also assigns considerable repair work to private yards. Many Naval vessels, as well as the commercial type ships of the Military Sea Transportation Service, are repaired in private shipyards. During the calendar year 1951, more than $86,000,000 was spent for overhaul of Naval and MSTS vessels under contract in private shipyards. The figure for 1952 is expected to run as high or higher.

Our 1952 program has stirred considerable shipyard activity in all parts of the country. Both large and small yards are participating. But it now appears that the 1953 program will be a modest one, perhaps only a third of the 1952 program in money value. Therefore, we cannot anticipate further expansion of our shipbuilding procurement base, and in fact we shall not be able to sustain it. However, we feel that the contracts we have awarded thus far have put the major operating yards in a position to give us a good and greatly improved mobilization base.

It has been our practice and policy to award contracts only to yards already in operation. We have not encouraged reopening of idle yards on the expectation of Naval work. The yards now performing our work were going concerns at the time the current naval ship-

building program began about two years ago. We hope that enough Naval work will be available to supplement their commercial work in order to sustain them at their present level. Obviously, this will require an increased flow of commercial work.

Now, in review: I have pointed out how this industry operates at a disadvantage in competition with foreign operators. The Merchant Marine Act of 1936 was intended to be part of a bold and daring plan to restore American merchant ships to their rightful place in world commerce. It set up a system of subsidies for building and operating ships, designed to bridge the gap between the costs of American and foreign building and operation. This Act resulted in the start of considerable construction just prior to World War II. The industry which had been flat on its face was enabled to rise to one knee, at least, before undertaking its part in the tremendous task of saving the nation in World War II.

The years since the war have shown that the Act of 1936 has not been able to do the job for which it was intended. The subsidy laws have run up against so much confusion and inconsistency of interpretation that they no longer offer inducement sufficient to get ships built. In 1947 the government did its best to urge private operators to build new passenger vessels for the depleted American passenger fleet under the terms of the Act. The result was contracts for only six vessels, three of which have been taken over by the Government, thus leaving only three to be added to our Merchant Marine.

It is clear that the national interest demands a program that will encourage private enterprise to build passenger ships and cargo ships in an orderly regular manner that will ensure a stable industry and will eliminate the nightmare of block obsolescence now facing our Merchant fleet as well as our Navy.

This program would be costly, but it is a cost that we can well afford when we consider what is at stake. When viewed in the light of our responsibilities of world leadership and our national program of subsidies for other purposes, the cost of an adequate merchant marine takes on minor proportions.

During the 16 years since the enactment of the Merchant Marine Act of 1936, shipbuilding subsidies have cost the government $377,000,000.

Yet, to support the price of Irish potatoes alone, the Federal Government spent more than $477,000,000 from late 1933 until 1951. Supporting the price of peanuts came to $83,000,000 during the same period and supports for grain sorghum totaled $38,000,000. During this period, the total loss to the government through support of agricultural prices was more than $981,000,000. This does not include the wartime program of direct subsidies, which cost more than two billion dollars.

During the average peacetime year, we have spent nearly three times as much money to support agricultural prices as we have spent to encourage the building of an adequate, up-to-date merchant marine.

I do not mean to imply disapproval of the farm support program. A healthy agriculture is certainly necessary to the country—but so is a healthy Merchant Marine. We must preserve a balance. A diminishing Merchant Marine does not appear consistent with an expanding interest and commitment in international affairs.

I am not asserting that by spending money we can insure security in any field. But I am stating that the American Merchant Marine, and the shipbuilding and repair industry which is a part of it, are vital to the existence and prosperity of this country. They are basic national resources, for peace and for war. As such, their needs must be considered in any program that seeks to

insure the long-range welfare and security of our nation.

In a world whose surface is 70% water, and always will be, and wherein American influence and trade must predominate, seapower is not only necessary, it is mandatory. So today, more than ever before, we should heed the formula of Admiral Mahan: "Sea power equals naval vessels plus bases plus merchant vessels." Thus the state of the Nation today, and always is tremendously affected by the condition of its shipping and shipbuilding!

THE NUMERICAL SOLUTION OF UNSTEADY-STATE HEAT CONDUCTION PROBLEMS BY THE METHOD OF CRANK AND NICOLSON

GEORGE LEPPERT

THE AUTHOR

served in the United States Navy from 1943 to 1946 and is a qualified submarine officer. Received B.S. degree in Mechanical Engineering from the University of Wisconsin in 1947, and the M.S. degree from Illinois Institute of Technology in 1952. Since 1950 he has been a member of the Heat Power Section of the Reactor Engineering Division at Argonne National Laboratory, engaged in the design of a nuclear power plant for submarine propulsion.

The numerical solution of unsteady-state heat conduction problems has been largely limited, until recently, to the method described by Emmons and Dusinberre.[1] This numerical procedure, although of great utility, in many cases requires a prohibitive amount of computation. The reason for this difficulty is that, for a particular set of physical circumstances, there is a definite upper limit placed on the size of time step permitted. If this limit is exceeded, the solution may become unstable and fail to converge. It is the purpose of this paper to review briefly a different method of numerical solution which is not subject to the same limitation on the size of the time step, and to indicate how the method may be applied in the engineering analysis of a few typical transient heat transfer problems. Although this discussion is primarily concerned with heat transmission, it is believed that this mathematical method will prove to be useful to workers in other fields of engineering design, such as for fluid flow calculations and for the study of electrical transients.

As an introduction to the subject, we may consider the equation for one-dimensional heat flow through a homogeneous solid without internal sources or sinks:

$$\frac{\partial \theta}{\partial t} = \alpha \frac{\partial^2 \theta}{\partial x^2} \tag{1}$$

The Emmons-Dusinberre procedure results in the replacement of these partial derivatives by certain finite difference ratios, so that at a point in the body, the difference equation would be

$$\frac{\theta_{m,n+1} - \theta_{m,n}}{\Delta t} = \alpha \frac{\theta_{m+1,n} - 2\theta_{m,n} + \theta_{m-1,n}}{(\Delta x)^2} \tag{2}$$

where $\theta_{m,n}$ refers to a temperature excess at a point $m\Delta x$ distant from the point $x = 0$ and $n\Delta t$ later than the time chosen as zero time. Throughout this discussion Δx and Δt denote finite increments of length and time, respectively, while α represents the thermal diffusiv-

ity of the material. Equation (2) may be solved for the future temperature at a point:

$$\theta_{m,n+1} = \frac{\theta_{m-1,n} + (M-2)\theta_{m,n} + \theta_{m+1,n}}{M} \tag{3}$$

where $M = \dfrac{(\Delta x)^2}{\alpha \Delta t}$

Now we may write one such equation for every point in the space subdivision and, with a set of initial temperatures $\theta_{m,o}$ $(m = 1,2,3, \ldots, p)$, the later temperature distributions $\theta_{m,1}$, $\theta_{m,2}$, ... , $\theta_{m,n}$ may be computed for each of the p points. It is seen that although this results in p algebraic equations in p unknowns, each equation is explicit in one future temperature, with everything else in each equation being known. Hence, the solution of the equations at each time step is simple and fairly rapid, although the number of time steps may be very large, as will be shown later. However, it is well known[2] that these equations may be expected to oscillate unless $M \geq 2$, so if the space subdivision is small, the amount of computation with this method may prevent its use. Incidentally, the often-used Binder-Schmidt graphical analysis is just a special form of this method with M=2. This results in each future temperature becoming a simple average of the present temperatures at adjacent points. Equation (3) becomes, with M=2,

$$\theta_{m,n+1} = \frac{\theta_{m-1,n} + \theta_{m+1,n}}{2}$$

To insure a stable solution, however, one usually chooses a somewhat larger value than two.

J. Crank and P. Nicolson[3] have described a different method of evaluating numerical solutions of an equation of the type

$$\frac{\partial \theta}{\partial t} = \alpha \frac{\partial^2 \theta}{\partial x^2} + \frac{q'''(x)e^{-A/\theta}}{\rho c} \tag{4}$$

where

$q'''(x)$ = known heat generation rate per unit volume at some reference temperature. This rate is seen to depend on location.

ρ = density of the solid.

c = heat capacity of the solid.

A = a constant.

(It will be seen that since the heat generation term in equation (4) is dependent on the temperature, no additional difficulty is introduced by temperature dependence of the density and/or heat capacity.) The equation (4) is subject to a given initial temperature distribution throughout the body and to heat transfer coefficients at both surfaces which are assumed to be known functions of the temperature excesses at those surfaces.

The method consists of replacing the space and time derivatives at the point $[m\Delta x, (n + \frac{1}{2})\ \Delta t]$ by finite difference ratios as follows:

$$\frac{\theta_{m,n+1} - \theta_{m,n}}{\Delta t} = \frac{\alpha}{2(\Delta x)^2}\left[\theta_{m-1,n+1} + \theta_{m+1,n+1} + \theta_{m-1,n} + \theta_{m+1,n} - 2(\theta_{m,n+1} + \theta_{m,n})\right] + \frac{q'''(m\Delta x)}{\rho c} e^{-A/\theta_{m,n+\frac{1}{2}}} \tag{5}$$

This equation may be rearranged to

$$\theta_{m,n+1} = \frac{\theta_{m-1,n+1} + \theta_{m+1,n+1} + \theta_{m-1,n} + \theta_{m+1,n} + 2\theta_{m,n}(M-1)}{2(M+1)} + \frac{M}{(M+1)} \frac{\Delta t}{\rho c} q'''(m\Delta x) e^{-2A/\theta_{m,n+1} + \theta_{m,n}} \tag{6}$$

It is seen that this equation, as with (3), above, leads to as many simultaneous equations as there are points in the physical subdivision. However, unlike those previously described, these may not be solved explicitly for each future temperature, because the heat generation term is also dependent upon the future (unknown) temperature. In a numerical example, the authors show that the equations may be solved fairly readily at each time step by an iteration procedure.

The importance of this method lies in the tendency of the finite-difference substitution which Crank and Nicolson used to damp out errors, i.e., the solution is stable even when the M-values are less than two. Although the general proof of this contention apparently does not exist, those authors have demonstrated this fact for the simple case without heat generation, and empirical corroboration is available for a number of other cases.

To illustrate the use of the method, and to compare its results with those of the explicit-type numerical solution, we may select the simple example of one-dimensional heat conduction in a homogeneous solid, which is described by the partial differential equation (1). Suppose the heated body is two feet in length, of unit area perpendicular to the direction of heat flow, and initially at the same temperature throughout; i.e., $\theta_{m,o} = 0$ for all m. Let the temperature of both end faces be raised suddenly and held constant at $\theta_{10,n} = 100$ F. for all $n>0$. The analytical solution to this problem is given by Carslaw and Jaeger[4] and is shown for comparison by the curves in Figure 1.

Instead of obtaining the finite difference equations from equation (1) by the substitutions shown in equations (2) and (5), let us use the physical reasoning employed by Dusinberre. This method of derivation is much more satisfying, and it also gives a valuable insight into the mechanics of the techniques which we are studying.

Figure 2 shows one-half of the body just described, from the plane of symmetry to one heated face. Subdivide as shown, with half-blocks of length $\frac{\Delta x}{2}$ at each surface, and with interior blocks of length Δx. Write a heat balance on the first half-block, shown in Figure 2.

heat stored per unit time = heat from block 1

$$\frac{\rho c \Delta x}{2\Delta t}\left(\theta_{o,n+1} - \theta_{o,n}\right) = \frac{k}{\Delta x}\left(\theta_{1,n} - \theta_{o,n}\right)$$

or
$$\theta_{o,n+1} = \frac{(M-2)\theta_{o,n} + 2\theta_{1,n}}{M} \qquad (7a)$$

Similarly, a heat balance on the first full block would be expressed as follows:

$$\frac{\rho c \Delta x}{\Delta t}\left(\theta_{1,n+1} - \theta_{1,n}\right) = \frac{k}{\Delta x}\left(\theta_{2,n} - \theta_{1,n}\right) - \frac{k}{\Delta x}\left(\theta_{1,n} - \theta_{o,n}\right)$$

or
$$\theta_{1,n+1} = \frac{\theta_{o,n} + (M-2)\theta_{1,n} + \theta_{2,n}}{M} \qquad (7b)$$

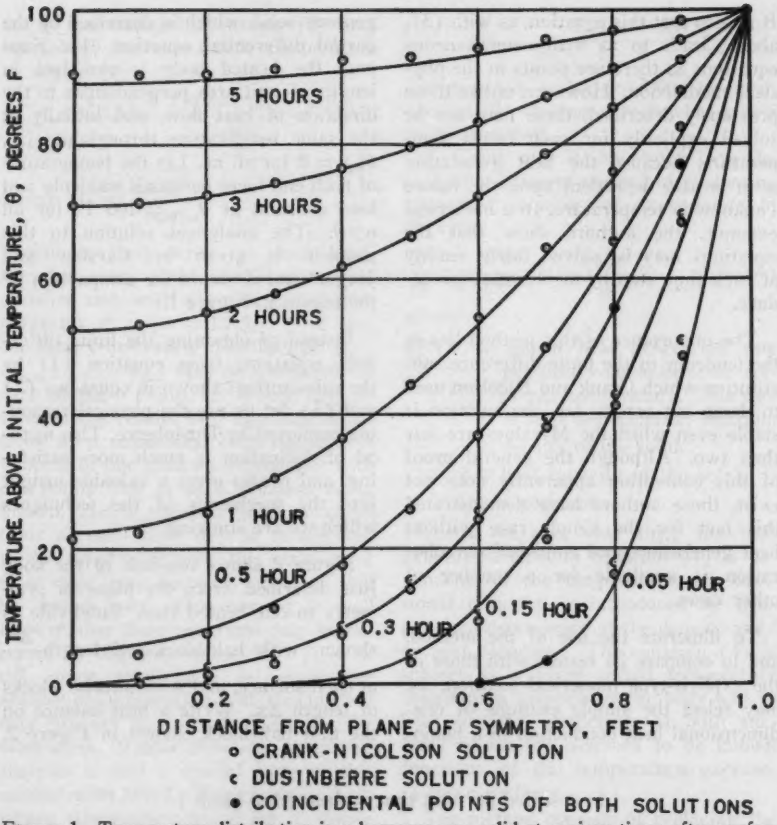

FIGURE 1—Temperature distribution in a homogeneous solid at various times after surface temperature is increased by 100 F.

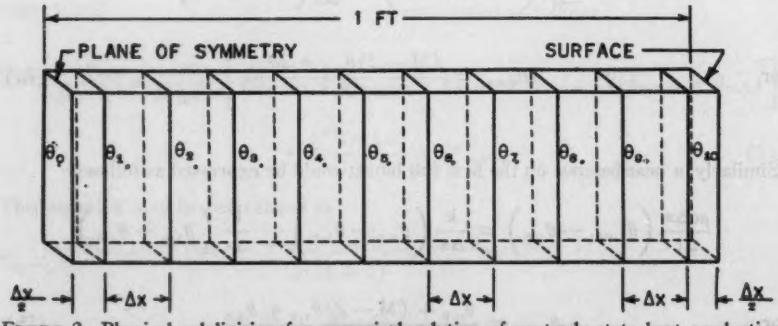

FIGURE 2—Physical subdivision for numerical solution of unsteady-state heat conduction problem in a homogeneous solid.

and so on for each of the future temperatures $\theta_{2,n+1}$, $\theta_{3,n+1}$, ... , $\theta_{9,n+1}$. ($\theta_{10,n}$ is held at 100 F. throughout the problem.) These equations are seen to be identical with those obtained by the finite difference substitution into equation (1) (of equation (3)).

If $\alpha = 0.2$ ft²/hr, and we choose $M = 4$, it is seen that $\Delta t = \frac{(\Delta x)^2}{M\alpha} =$ 0.0125 hrs. Hence, if we wish to find the temperature distribution after five hours, we must carry out the solution of these ten simultaneous equations, which are explicit in the future temperatures, for $n = \frac{5}{0.0125} = 400$ steps. This calculation has been performed for the first 0.30 hrs., and the temperatures obtained are shown on the first three curves of figure 1.

To see the physical significance of equation (6), as applied to this problem, let us again set up a heat balance on the first half-block in Figure 2:

heat stored per unit time = heat from 1

$$\frac{\rho c \Delta x}{2\Delta t}\left(\theta_{o,n+1} - \theta_{o,n}\right) = \frac{k}{\Delta x}\left(\theta_{1,n+1/2} - \theta_{o,n+1/2}\right) \tag{8}$$

In this heat balance, a different set of temperatures is used in the conduction terms. Instead of expressing the temperature gradient as a function of the temperatures at the beginning of the time interval, it is expressed in terms of average temperatures over the interval. That is, $\theta_{m,n+1/2}$ is evaluated as the arithmetic mean of present and future temperatures:

$$\theta_{m,n+1/2} = \frac{\theta_{m,n+1} + \theta_{m,n}}{2} \tag{9}$$

Then the above equation (8) becomes:

$$\theta_{o,n+1} = \frac{\theta_{1,n+1} + \theta_{1,n} + \theta_{o,n}(M-1)}{(M+1)} \tag{10a}$$

On the first full block, the heat balance is:

$$\frac{\rho c \Delta x}{\Delta t}\left(\theta_{1,n+1} - \theta_{1,n}\right) = \frac{k}{\Delta x}\left(\theta_{2,n+1/2} - \theta_{1,n+1/2}\right) - \frac{k}{\Delta x}\left(\theta_{1,n+1/2} - \theta_{o,n+1/2}\right)$$

or

$$\theta_{1,n+1} = \frac{\theta_{o,n+1} + \theta_{2,n+1} + \theta_{o,n} + \theta_{2,n} + 2\theta_{1,n}(M-1)}{2(M+1)} \tag{10b}$$

Similarly,

$$\theta_{2,n+1} = \frac{\theta_{1,n+1} + \theta_{3,n+1} + \theta_{1,n} + \theta_{3,n} + 2\theta_{2,n}(M-1)}{2(M+1)} \tag{10c}$$

and so on. These equations are equivalent to equation (6).

As with the previous type of equation, we obtain ten simultaneous algebraic equations with the ten unknown temperatures $\theta_{0,n+1}$, $\theta_{1,n+1}$, ... , $\theta_{9,n+1}$. In this case, however, each equation contains either two or three of these unknown $\theta_{m,n+1}$ terms, so the solution for one time step appears somewhat more difficult. Still, the process is simple, because one sees that equation (10a) always represents a known relation between the unknowns, $\theta_{0,n+1}$ and $\theta_{1,n+1}$. This relation, substituted in equation (10b), produces a similar expression between $\theta_{1,n+1}$ and $\theta_{2,n+1}$. This substitution may be repeatedly made through the equation for $\theta_{8,n+1}$, which yields the relation between $\theta_{8,n+1}$ and $\theta_{9,n+1}$. However, the last equation, written for $\theta_{9,n+1}$, contains as unknowns only $\theta_{8,n+1}$ and $\theta_{9,n+1}$, so these quantities may then be determined, and, from them, all the rest of the $\theta_{m,n+1}$ values at that time step.

For the same initial conditions as before, then, this set of ten equations has been solved simultaneously at each of thirteen time steps: t = 0.025, 0.05, 0.10, 0.15, 0.20, 0.25, 0.30, 0.40, 0.50, 1.0, 2.0, 3.0, and 5.0 hours. The results of this solution are shown in Figure 1.

In the early part of the problem, where the time steps are comparatively small (yet larger than those of equations 7), the solution is simplified by the presence of many zero temperature excesses, as shown by Figure 1, and also by the fact that the solution is simpler when the same size of time interval is used for more than one step of the solution. Even in the later part of the problem, however, it was found that about twenty minutes' computing time with a desk machine was sufficient to solve the ten equations. In the last step, for instance, from t = 3.0 to 5.0 hours, this is to be compared with the much more tedious solution of the ten explicit equations (7) through 160 time steps to cover the same time interval.

Another advantage to the Crank-Nicolson method, incidentally, is that the steps may be chosen large enough to keep round-off error negligible, whereas in a solution where one must keep $M \geqq 2$ to prevent oscillation, it is sometimes found that this error becomes of comparable magnitude to the temperature change from one time step to the next. In order to understand the nature of this round-off error in numerical calculations, let us examine the analytical solution represented by the curves of Figure 1. Between the last two curves, from t = 3.0 to 5.0 hours, the temperature θ_7 changes by a little less than 8F. As mentioned previously, the time interval between these curves is covered by 160 time steps in the Emmons-Dusinberre equations. If the computation is being carried out to the nearest tenth of a degree, it is possible to acquire an error of nearly one-twentieth of a degree by "rounding off" a temperature a single time. However, the actual solution changes by an average of only $\frac{8}{160}$, or one-twentieth of a degree per time step, so the round-off error in one time step is sufficient to obscure the temperature change being sought. If one is able to take larger time steps, of course, this difficulty is avoided.

Now that the general features of the Crank-Nicholson Method have been described, it might be useful to describe its application to a more complex problem. Suppose that a long composite metal plate, Figure 3, generates heat in the interior, as, for instance, by a nuclear reaction or by the passage of a large electrical current. Suppose, further, that at the surfaces a different metal is bonded to this current-conducting metal, as indicated in Figure 3, and that no heat is generated in this clad-

ding material. The exposed surface of the latter metal is cooled by the forced convection of a fluid, so that the surface heat transfer coefficient, h, is primarily a function of the velocity of this fluid. It is assumed that one knows the steady-state values of heat generation, fluid temperature, heat transfer coefficient, and properties of both metals, such as density, heat capacity, and thermal conductivity. From this data, of course, one may calculate the steady-state temperature distribution which is needed to start the transient solution.

The unsteady-state problem starts when the power, fluid flow rate, or fluid temperature is changed by external factors. For example, the heat generation rate might be varied in a given fashion with time. Furthermore, this might be accompanied by specified variations in the coolant flow rate and temperature.

The first step in the solution is to choose a reasonable physical subdivision through the plate, as shown in Figure 3. Because of symmetry, only half the plate need be considered. Also, since the area normal to the heat flow is constant, it may be taken as unity. Writing a heat balance for each block in the subdivision, as for equation (8), we obtain six simultaneous equations:

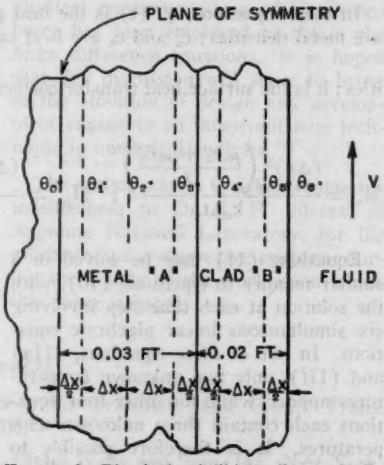

FIGURE 3—Physical subdivision for numerical solution of unsteady-state heat conduction problem with internal heat generation in a long, thin plate.

$$\theta_{0,n+1} = \frac{\theta_{1,n} + \theta_{1,n+1} + \theta_{0,n}(M_a - 1)}{M_a + 1} + \frac{M_a q'''(t)\Delta t}{\rho_a c_a (M_a + 1)} \tag{11a}$$

$$\theta_{1,n+1} = \frac{\theta_{2,n} + \theta_{2,n+1} + \theta_{0,n} + \theta_{0,n+1} + 2\theta_{1,n}(M_a - 1)}{2(M_a + 1)} + \frac{M_a q'''(t)\Delta t}{\rho_a c_a (M_a + 1)} \tag{11b}$$

$$\theta_{2,n+1} = \frac{\theta_{3,n} + \theta_{3,n+1} + \theta_{1,n} + \theta_{1,n+1} + 2\theta_{2,n}(M_a - 1)}{2(M_a + 1)} + \frac{M_a q'''(t)\Delta t}{\rho_a c_a (M_a + 1)} \tag{11c}$$

$$\theta_{3,n+1} = \frac{M_{ba}(\theta_{2,n+1} + \theta_{2,n}) + M_{ab}(\theta_{4,n+1} + \theta_{4,n}) + (2M_{ab}M_{ba} - M_{ba} - M_{ab})\theta_{3,n} + \dfrac{2M_{ba}M_{ab}q'''(t)\Delta t}{\rho_a c_a + \rho_b c_b}}{2M_{ab}M_{ba} + M_{ba} + M_{ab}} \tag{11d}$$

$$\theta_{4,n+1} = \frac{\theta_{5,n} + \theta_{5,n+1} + \theta_{3,n} + \theta_{3,n+1} + 2\theta_{4,n}(M_b - 1)}{2(M_b + 1)} \tag{11e}$$

$$\theta_{5,n+1} = \frac{N(\theta_{6,n+1} + \theta_{6,n}) + \theta_{4,n+1} + \theta_{4,n} + (M_b - N - 1)\theta_{5,n}}{M_b + N + 1} \tag{11f}$$

In these equations, $q'''(t)$ is the heat generation rate per unit volume; ρ_a and ρ_b are metal densities; c_a and c_b are heat capacities; k_a and k_b are thermal conductivities; h is the surface heat transfer coefficient; $M_a = \frac{(\Delta x)^2}{\alpha_a \Delta t}$; $M_b = \frac{(\Delta x)^2}{\alpha_b \Delta t}$;

$$M_{ab} = \frac{(\Delta x)^2 \left(\frac{\rho_a c_a + \rho_b c_b}{2} \right)}{k_a \Delta t}; \quad M_{ba} = \frac{(\Delta x)^2 \left(\frac{\rho_a c_a + \rho_b c_b}{2} \right)}{k_b \Delta t}; \text{ and } N = \frac{h \Delta x}{k_b}$$

Equations (11) may be solved in a similar manner to equations (10), with the solution at each time step involving six simultaneous linear algebraic equations. In two of these equations, (11a) and (11f), only two unknown temperatures appear, while the other four equations each contain three unknown temperatures. It is therefore possible to eliminate the unknowns successively so that an iterative solution at each time step, as in the example of Crank and Nicolson (equation (6)), is not required.

Before leaving this example, it is interesting to consider the application of the Emmons-Dusinberre type of solution for a transient of one or more hours' duration. If the thermal diffusivity is 0.2 ft²/hr, and Δx is 0.01 ft, the minimum length of a time step in the interior for $M = 2$, is $\frac{(0.01)^2}{(2)(0.2)} = 0.00025$ hrs., so that we must have at least 4000 time steps per hour. At the surface, the criterion for stability has been shown to be $M \geq 2N + 2$, where $N = \frac{h \Delta x}{k}$. For $h = 1000$ B/hr ft²F, this would require $M \geq 4$, which represents a minimum of 8000 time steps per hour. It is seen, then, that this method would not offer a practical solution unless one were willing to set it up on a large digital computing machine.

The Crank-Nicholson Method, on the other hand, has been shown to approximate the analytical solution of Figure 1 with M-values as low as one-twentieth, which corresponds to about 100 time steps per hour. Actually, even lower M-values are acceptable if the parameters are changing gradually instead of in the discontinuous fashion of the example represented by Figures 1 and 2. For problems with heat generation in a thin plate, similar to the one just described, the author has compared solutions of the explicit equations, carried out for thousands of time steps by a large punched-card machine, with those obtained by the Crank-Nicolson Method with desk-type calculating machines and with only a small fraction of the number of time steps. The temperatures obtained were quite comparable, differing by considerably less than the usual errors which one anticipates from the use of the accepted convective heat transfer and fluid friction correlations.

Finally, it may be pointed out that the equations (11) are applicable and may be handled satisfactorily even if some or all of the parameters and physical properties are dependent upon temperature as well as upon time. For instance, the heat generation might very well depend in a known way upon the temperature of the inner metal; the heat transfer coefficient might depend upon the surface and fluid temperatures; and the thermal conductivities of the two metals might depend upon their temperatures. In this case, the six simultaneous equations must be solved by iteration at each time step. This does not prove to be a serious complication, however, as is pointed out in the Crank-

Nicolson paper for the problem they considered (equation (6)).

To summarize, we have reviewed a new mathematical method proposed by Crank and Nicolson for the solution of partial differential equations of the heat conduction type, and we have compared this with the Emmons-Dusinberre and with the Binder-Schmidt Methods. The advantages of this new technique have been pointed out and illustrated by examples in which, instead of purely mathematical derivations, the type of physical reasoning advanced by Dusinberre has been employed to obtain the finite difference equations. It is hoped that this discussion will serve to bring to the attention of design and development engineers an important new technique in numerical analysis.

The author wishes to acknowledge his indebtedness to Dr. J. P. Silvers, of Argonne National Laboratory, for his guidance and constructive criticism during the course of the work upon which this paper is based.

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NEW ADMIRALTY DIESEL ENGINES

ACKNOWLEDGMENT

This is a staff article appearing in the February 1952 issue of "The Motor Boat and Yachting."

A new series of Diesel engines for propulsion and auxiliary purposes in the Royal Navy has been put into production. It covers a power range of 750-2000 h.p. Work on this series began in 1945 to meet the need for motors having a better power/weight ratio than those then available from commercial sources.

Detailed design was undertaken by the Engineering Department of H. M. Dockyard, Chatham, in conjunction with the Admiralty Engineering Laboratory, West Drayton, and the manufacture of a prototype 16-cylinder version was commenced at Chatham at the end of 1948.

The engine so developed was called the Admiralty Standard Range I (A.S.R.I.) engine, and designs have now been completed for the 16- and 12-cylinder "V" and the 8- and 6-cylinder "in line" versions in both supercharged and unsupercharged forms.

Numbers of these engines are now being produced at Chatham Dockyard and by various well-known firms for general application in the Navy.

The main particular of A.S.R.I engines are given in the appended table.

Features common to all engines in the series are direct injection with four-valve cylinder heads, all-welded steel frame, chromium plated cylinder liners, copper-lead main and big-end pre-finished thin shell bearings (big-end bearings for "V" units being of the fork and blade type), fresh water cooling, and hydraulic governors on both propulsion and auxiliary engines.

Particular attention has been paid to interchangeability of all replaceable parts, including similar crankshafts and bedplates for "in line" and "V" versions of equivalent number of crank throws. Accessibility and ease of maintenance have, it is stated, been borne well in mind in preparing the design.

	16 V.T.S.	*12 V.T.S.*	*8 L.T.S.*	*6 L.T.S.*
Bore	....	9¾"	....	....
Stroke	....	10½"	....	....
Maximum r.p.m.	....	1,000	....	....
Maximum b.h.p. supercharged	2,000	1,500	1,000	750
Maximum b.h.p. unsupercharged	1,400	1,050	700	525
Dry weight (lb.) (including engine mounted auxiliaries)	36,000	29,000	24,000	19,000
	ft. in.	ft. in.	ft. in.	ft. in.
Length	16 10	14 0	16 6	14 0
Breadth	5 4	5 4	4 3	4 3
Height	8 7½	8 6	8 7	8 6

UNITED STATES NAVY RUBBER CRAFT

COMMANDER ARTHUR CLIFTON BUSHEY, JR., USN

THE AUTHOR

COMMANDER BUSHEY is stationed in the Bureau of Ships, Contract Design (Hull). He was commissioned 28 February 1930 as a Lt. (J.G.)—Construction Corps, U. S. Naval Reserve. On 7 October, 1940 he was called to active duty and assigned to the Design Section of the Norfolk Navy Yard. To aid the training of personnel in that rapidly expanding organization, he organized the University of Virginia Norfolk Extension for Naval Architecture, teaching in the evenings for two years. He was next assigned to the Pacific. He holds the Bronze Star Medal. The late King of England appointed him an O.B.E. in recognition of his services as officer in charge of shipbuilding and ship repairs in New Zealand and commanding officer of the U. S. forces in Auckland.

It took nearly 300 years after the first white man saw South American Indians playing with a "rubber" ball before the use of rubber became widespread. It was then utilized for erasers; hence the name "rubber"—to rub out pencil and ink marks.

Nearly all of the rubber for the past half of a century has been obtained from tapping the so-called rubber trees (Hevea brasiliensis), growing in large tropical plantations, for a milk-like substance known as latex. Hevea Latex produces the largest quantity and the best quality of caoutchouc (pure rubber).

The uses of rubber and synthetic rubber in various forms by the military services of all nations are almost endless: tires, airships, barrage balloons, pontoons, salvage bags, fuel storage tanks, submerged refueling bags, decoys and camouflage equipment, life preservers, boats, exposure suits, de-icers for aircraft wings, self-sealing fuel tanks, etc.

The equipment as we use it can generally be broken down into two categories, the mechanical goods items such as tank treads, hose, belting, mats, decking, etc., and the coated fabrics items such as airships, balloons, boats, etc.

Probably the first manufacturer of rubber goods in this country was the India Rubber Company of Roxbury, Massachusetts, who, in 1832, made rubber bottles, overshoes, mackintoshes (named after Charles Macintosh who, in 1823, manufactured the first waterproof cape in England. This was highly unsuccessful as the material hardened like armor plate in winter and softened and fell apart in summer). In 1839, Charles Goodyear in New Haven, Connecticut, developed vulcanized rubber by heating rubber with sulphur. The old temperature difficulties were overcome by this process and, in addition, a more elastic, stronger and more durable material became available for our people of more than a century ago. Further developments followed, combining rubber with

15-Man Capacity Boats.
Inflatable Type with Double Canopy 1951.
(Packs in Carrying Case 5′-0″ x 2′-9″ x 2′-2″. Weight 200 Pounds.)

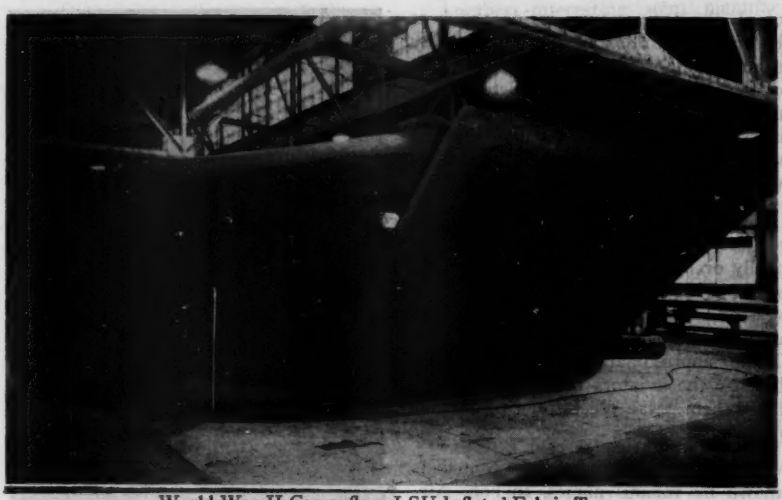
other materials, by blending and, later, by coating and taking advantage of their reinforcing properties, to obtain a wide variety of desired qualities in the end products; and more recently, the economies and improvements made possible through utilization of scrap rubber.

The increasing use of aircraft was responsible for the military services and a few rubber companies, some no longer in existence, undertaking a continuing program of design and development of better inflatable equipment for rescue at sea. This program is still going on.

With the advent in 1930 of efficient automatic inflation devices, a great step forward was made. This was accomplished by the use of compressed carbon dioxide (CO_2) in a steel cylinder discharging through a valve and manifold into both compartments of the boat. (Nitrogen was added to help gas discharge at —65° F.).

With this development, the self-inflating rubber boat became a practical reality and its use increased tremendously as its ability to "bring 'em back alive" was attested to by those who were saved.

In 1938, the life boat was modified for landing amphibious troops and was able to successfully land troops across coral reefs and through surf. These boats were fitted with outboard motors and machine gun mounts. Early in 1940 development work on pontoons was started. These pontoons were oval shaped, rubberized fabric rafts, fitted with "D" rings to which a treadway was secured and when placed side by side across a river, allowed the heaviest tanks to cross on a bridge of air. These pontoons are readily deflated and the



World War II Camouflage LSU Inflated Fabric Type.

disassembled bridge becomes relatively easy to transport.

During World War II a large variety of camouflage items were made of coated fabric, including aircraft, practically all mechanized and mobile equipment and landing craft up to the LCT or LSU, which is over 100 feet long. This, together with a model of a PT, practically drove the people involved in the program to the verge of insanity.

British observers working with our people on this equipment stated that the Suez Canal was closed by the British for a 24-hour period when an unidentified submarine was reported near the entrance. Actually the camouflage people had built the superstructure of a submarine and had it under test. It was a secret project and the secret was apparently too well kept.

The critical crude rubber situation about 1942 precluded its use for shipboard boats, pontoons and camouflage items; so the rubber industry did a remarkable job in the conversion to synthetics, the most satisfactory one being Neoprene. Neoprene is still used and is superior to the crude rubber compounds in some respects (oil resistance—abrasion).

During the war (and to some extent even now) U. S. and British naval vessels were fitted with floater nets (hard cellular rubber floats secured together by ropes through their horizontal centers), and balsa floats. Both of these are practically useless in cold water and resulted in terrifically high personnel losses in British ship sinkings, the impact of which was felt later when new ships were added to their fleets. Since the war, considerable work has been done to develop an inflatable boat for shipboard use which would be of value in cold water. The 15-man capacity boats presently being procured for shipboard use are fitted with a double canopy, which has about a 4-inch air space between the inner and outer layers. Tests of these boats were conducted in water at a temperature of 40° F. with an air temperature of 47° F. Ten men boarded the boat from the water after being immersed from 1 to 11 minutes, and within a short time (about 30 min-

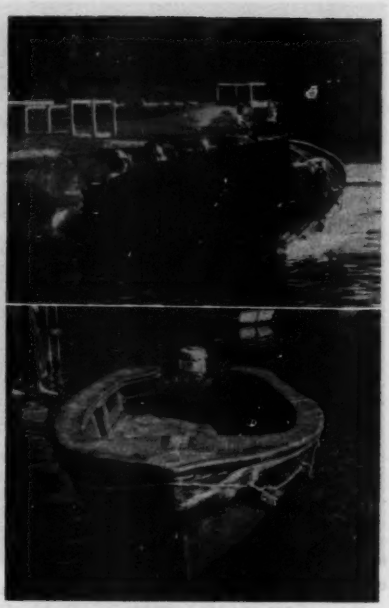
utes) the temperature inside the boat had built up to 87° F. and the humidity to almost 100%, just from the body heat of the occupants. This completely upset the popular fallacy that a person immersed in cold water for five minutes was pretty sure to succumb no matter what treatment was given.

The use of inflatable life preservers made from coated fabrics, and inflated orally or by CO_2, was initiated for aviators. The life preserver is worn as a vest and when inflated will support an unconscious man in an upright position. These approximately date themselves because the aviators immediately christened them "Mae West" for two obvious reasons.

The latest material being coated is known by various names, In-Fab, Air-Mat, Rigid-Aire, depending on the company using it; a neutral name and the one in use by the U. S. Navy is denoted Inflatable Mat Fabric. This material is woven on plush weaving looms and can be readily made from ⅛" thick to 3" thick of heavy tire cord or gossamer thin nylon. Greater thicknesses can be made for special applications.

For most applications the material is made from nylon for its strength characteristics and as condensation will occur, together with the possibility of water entering through the valves, it has been found to be infinitely better than mildewproof cotton fabric.

This material (Inflatable Mat Fabric) is difficult to coat because the double thickness of material, together with the pile threads going through the spreader rolls, tends to develop wrinkles. Although the finished coated material with a reinforcing layer of cloth top and bottom costs about $18.00 per yard (52" wide), the use of this product reduces the assembly time of the end item with a resultant reduced labor cost, which tends to balance the initial material cost. The smooth surface of this material, when inflated, permits the manufacture of items hitherto unobtainable. In camouflage items the proper shadow effects and contours are extremely important. For boat bottoms, it allows greater inflation pressures and stiffer bottoms.



Experimental 10-Man Inflatable Mat Boat. Weight 200 Pounds without Engine 1951.

The experimental 10-man inflatable mat boat is made of two thicknesses of this material throughout, the inner shell being inflated by one inflation system and the outer shell by another one. The boat complete (less engine) weighs about 200 pounds and when packed in its carrying case (coated fabric) measures about 6'-0" x 2'-3" x 2'-3". The inflated boat is approximately 18'-0" long by 6'-0" beam and with a—

10 hp. outboard motor and 10 men, made 6.5 MPH

14 hp. outboard motor and 10 men, made 7 MPH

25 hp. outboard motor and 10 men, made 12.5 MPH

10 hp. outboard motor and 2 men, made 12.5 MPH

14 hp. outboard motor and 2 men, made 17 MPH

25 hp. outboard motor and 2 men, literally started to fly

and operators requested flight pay if they were to emulate pilots.

Another boat, a six-man reconnaissance boat, has a two-ply bottom of this material under two 16" diameter tubes. This boat complete (less engine) weighs about 265 pounds. In its carrying case it measures approximately 4 ft. long, 3 ft. wide and 1'-6" high. The inflated boat is about 14 ft. long and 5 ft. wide, and with a 25-hp. motor and a six-man load made 15 MPH and with a 25-hp. outboard motor and two men aboard made 27 MPH.

Another interesting item manufactured from coated fabric is the salvage pontoon, one or more of which are secured to a sunken vessel by divers and at the proper time inflated by the rescue ship's air line and the friendly air again provides cheap buoyancy.

The wide variety of service items previously mentioned are manufactured from cotton or nylon fabrics, coated with rubber or neoprene, and range in weight in ounces per square yard from 1 to 100. Tensile strengths vary up to 1,000 pounds per square inch, depending upon the service requirements of the end item.

Other fabrics and coatings are used and the development of other materials and uses continues. One of great interest now is the installation of "bladder" fuel tanks on many small craft to save weight. These tanks should not be confused with the well known self-sealing tanks.

THE FUTURE OF THE FLYING BOAT

ACKNOWLEDGMENT

HENRY KNOWLER is the author of this Fifth Louis Blériot lecture, a condensed version of which was published in the March 14 and March 21, 1952 issues of "Engineering." The complete lecture was read before a meeting of the Association Francaise des Ingénieurs et Techniciens de l'Aéronautique in Paris on March 12, 1952.

During the three decades before the Second World War, the hull form of flying boats remained substantially the same, with a transverse main step a little aft of the center of gravity and a second step situated well aft. In the course of time, the second step was gradually reduced in width until finally the aft planing bottom terminated in a point. Until recently, it has been necessary to make the depth of the main step about 8 percent of the maximum beam to avoid instability. This was a major cause of drag. In early developments a change of angle between the fore and aft planing bottoms had been found to be necessary, and this has persisted up to the present time. An angle of about 8 deg. is usual, but 5 deg. to 11 deg. have been used. Other hull features which have persisted are the heavy turn-down of the bow chine in order to deflect spray, and the retention of a sharp chine throughout the length of the planing bottom.

Throughout the period which ended shortly before the Second World War, flying boats were in advance of landplanes in size and in their attainment of range. However, as a consequence of the large size of hull required, firstly, for buoyancy, and, secondly, in order to raise the wing and propellers above waves and heavy spray—and of the high drag associated with the contemporary boat form—the speeds attained fell behind those of landplanes. The Schneider Trophy racing aeroplanes, however, demonstrated the possibility of building high-speed water-based fighters.

Turning to contemporary flying boats, ocean-patrol boats are required for operation from areas which are not suitable for landplanes. Their primary duty is submarine search, but they are also used for locating and destroying enemy transports, convoy escort, shadowing enemy ships, air-sea rescue, etc. Long endurance, coupled with a large radius of action, is required. For search or convoy escort a slow speed at a relatively low altitude is necessary, although high cruising speed out to the scene of action is also desirable. The various duties of piloting, navigation, search and protec-

tion, call for a relatively large crew. A high standard of seaworthiness is required, so that the aircraft can alight on the open sea. Modern examples of this type of flying boat are the Glenn Martin Marlin P5M-1 and the Convair XP5Y-1.

The latter is the first example of hull design in which increased fineness ratio has been used. The aircraft weighs 138,000 lb. but can carry a considerable overload. It is fitted with four Allison T.40 coupled turbines, driving contra-rotating propellers, with a total equivalent horsepower of 22,000. On account of its low power loading it has a high top speed of 390 m.p.h.

It is hardly necessary to recall the honored place, in passenger transport development, in which the Short Empire boat stands. Although the original design dates back to 1935, the Solent IV, a derivative of the original Short C-class boat, is still in operation. The first Saunders-Roe Princess, designed as a civil flying boat, is nearing completion by the builders. It has a take-off weight of 330,000 lb., and is fitted with ten propeller-turbine engines of a total equivalent horsepower of 35,000, estimated to give it a high cruising speed of 380 m.p.h. at 40,000 ft. The cross-section of the hull, for pressurizing reasons, is a figure eight; consequently, the cabin space is arranged on two decks. This is the first flying-boat hull to be pressurized. The designed range is 5000 miles when carrying 105 passengers. The hull design embodies a number of new features. The main step, in plan, is semi-elliptical, and the aft bottom is faired away directly from the step without a break in depth. This has resulted in a useful reduction in air drag, while providing one of the most stable hull models so far tank-tested.

With the advent of the jet engine, which has removed the necessity for mounting the propeller and engine high above the water, a new type of fighter has become possible. The first and only example so far built is the Saunders-Roe S.R./A1, first flown in 1947. It does not include some recent developments in flying boat form which have made possible a further considerable reduction in drag. The S.R./A1 weighs 15,000 lb., and is fitted with a pair of early axial-flow jet engines, manufactured by Metropolitan-Vickers. The armament consists of four 20-mm. cannons.

There is a steady demand for amphibians for inter-island traffic, forestry patrol and river communications, naval communications and air-sea rescue. Probably the best example of the naval communication and air-sea rescue amphibian, although not the most recent, is the Vickers Supermarine Walrus, and its later version, the Sea Otter. Being exceptionally seaworthy, they were found to be very suitable during the Second World War for air-sea rescue. A recent civil amphibian is the Short Sealand, a passenger-carrying machine, suitable for inter-island traffic and river communication. It weighs 9100 lb. and is fitted with two de Havilland Gypsy Queen engines. It carries seven passengers and has a range of 220 miles.

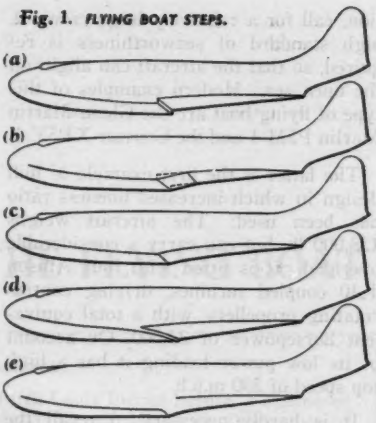
Up to and during the Second World War, flying-boat hulls were poor aerodynamically. From an analysis of the factors contributing to the increased drag of a hull over that of a streamline form, comprising a body of revolution based on the shape of the airship R.101, it was found that contemporary hulls had a relative drag factor of 1.43 to 1.51 times that of the streamline form. From wind-tunnel tests at high Reynolds numbers on representative hull bodies carried out by the National Advisory Committee for Aeronautics in America and the National Physical Laboratory in this country, together with certain additional information available, an analysis of drag increments for a typical hull of the form used during the Second World War, stated as a percentage increase in

drag relative to a body of revolution, is as follows:—cambering the basic streamline, 3 to 5 percent; adding fin and cabin, 2 to 3 percent; adding fore and aft planing bottoms, 8 to 9 percent; squaring the mid-body chines, 1 percent; turning down of bow chine, 1 to 8 percent; addition of main step, 20 to 38 percent.

The most serious cause of drag is the main step, the purpose of which is to break the water flow over the bottom of the hull at a point approximately where the normal pressure from the water diminishes and further aft changes to suction. Fig. 1 (*a*) shows a traditional form of step. A hull without a main step cannot leave the water because of this suction and it is necessary, therefore, to supply adequate air ventilation, in order to prevent the spray which leaves the step from rising and clinging to the aft planing bottom. A step which retracts in flight, as in Fig. 1 (*b*), has been tried, but the weight and difficulty of maintenance have ruled it out as a practical proposition.

Another solution used with considerable success has been to add a fairing behind the main step (Fig. 1 (*c*)), in elevation alone or by combining the fairing in elevation and plan. The drag of transverse steps, having straight fairings in elevation, decreases appreciably with fairing ratios above 3, the optimum fairing ratio being 9. At this value the total hull drag is approximately 20 percent above that of the basic body of revolution under conditions of minimum drag. With long straight fairings the natural ventilation of air along the break of the step must be supplied by other means. One method is to ventilate through air ducts to slots in the region of the step, with additional air ventilation slots some way down the aft planing bottom. Concave fairings are inferior to straight fairings and, although convex fairings are an improvement aerodynamically, they are unsuitable for hydrodynamic reasons.

Fig. 1. *FLYING BOAT STEPS.*



Applying the fairings with taper in plan, as well as elevation (Fig. 1 (*d*)) further improves the drag and, as the included angle of V is reduced, the drag due to the step decreases and virtually disappears when the angle is 10 deg. Unfortunately, satisfactory hydrodynamic properties have not so far been associated with small angles of V. A compromise in hull step fairing has been applied to the Saunders-Roe Princess (Fig. 1 (*e*)). This has a 6 to 1 fairing, springing from a semi-circular break in the bottom lines. The drag of this hull is approximately 15 percent greater than that of the basic body of revolution.

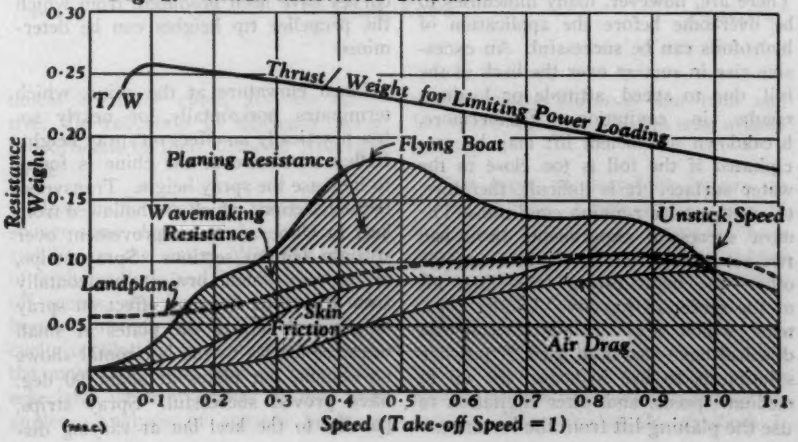
Fairings of greater length have been tested up to ratios of 20 or more, and forced ventilation has been used in order to overcome instability troubles. The improvement in air drag was, however, small. A certain amount of preliminary work has been done in the testing tank on the elimination of the step altogether by the use of forced ventilation. A deflection of 7½ percent of the jet-engine compressor delivery to a suitable pattern of air jets below a streamline body caused the body to run stably at satisfactory attitude and resistance.

It has hitherto been customary to make the forward planing bottom length 3 to 3½ times the maximum beam, corresponding to an over-all length/beam ratio of 6 to 7. Recent tank tests on a series of hulls of differing fineness ratio have led to an empirical relationship connecting length l and beam b where, for equivalent hydrodynamic behavior, $l^2 \times b =$ constant. Therefore the length-to-beam ratio can be varied and much higher beam loadings can be accepted. This has encouraged the projected use of hulls of greater fineness ratio, with corresponding low beam. In this way the drag increment due to the step can be reduced. For example, an increase in the forebody length-to-beam ratio from 3.5 to 6.5 results in a reduction in hull drag of 5 percent.

The basic streamline form can be chosen to suit the function of the aircraft, a high length-to-beam ratio being preferable. The depth of body will be decided either by the internal disposal of payload or by the necessity for providing adequate clearance of the power plant from spray. The planing bottoms, together with the keel and chines, are responsible for about 9 percent increase in drag over the basic body. This figure can be reduced a little by carrying the aft bottom to the sternpost, and, at the same time, the cambering of the basic streamline can be reduced. By careful design in the bow, so that the chines follow the air-flow pattern, the large increment of drag observed in one set of tests can be avoided. Alternatively, the drag cost for heavy bow turn-down can be eliminated completely by the use of spray plates which are retracted after take-off. It is also possible to make the mid-body chines follow the natural streamline flow for the mean cruising attitude. To summarize, the basic drag of the ideal streamline body need not be increased by more than 10 to 12 percent in converting it to a suitable hull form, compared with a 4- to 5-percent increase in converting the streamline form to a typical landplane fuselage.

The resistance during the take-off of a flying boat is the sum of quantities due to air drag, wave-making resistance, planing resistance and skin friction. Fig. 2 shows how the resistance varies with speed during the take-off run. The wave-

Fig. 2. WATER RESISTANCE DIAGRAM.



making resistance is greatest during the early part of the take-off but, being proportional to displacement, it reduces as the wing lift increases, and is of low value beyond the "hump." Planing resistance varies with $(\text{speed})^2$ and is directly proportional to area and attitude. At present, a refined form of flying boat has a resistance to weight ratio of 0.18. It is not expected that further important reductions can be made with a conventional type of hull. The skin friction component may be increased considerably by the impact of spray. If negative pressure is allowed to build up on the aft planing bottom the hull may ride at inefficient attitudes and additional lift is required to balance the downward suction. The take-off drag of a landplane is shown in Fig. 2 for comparison.

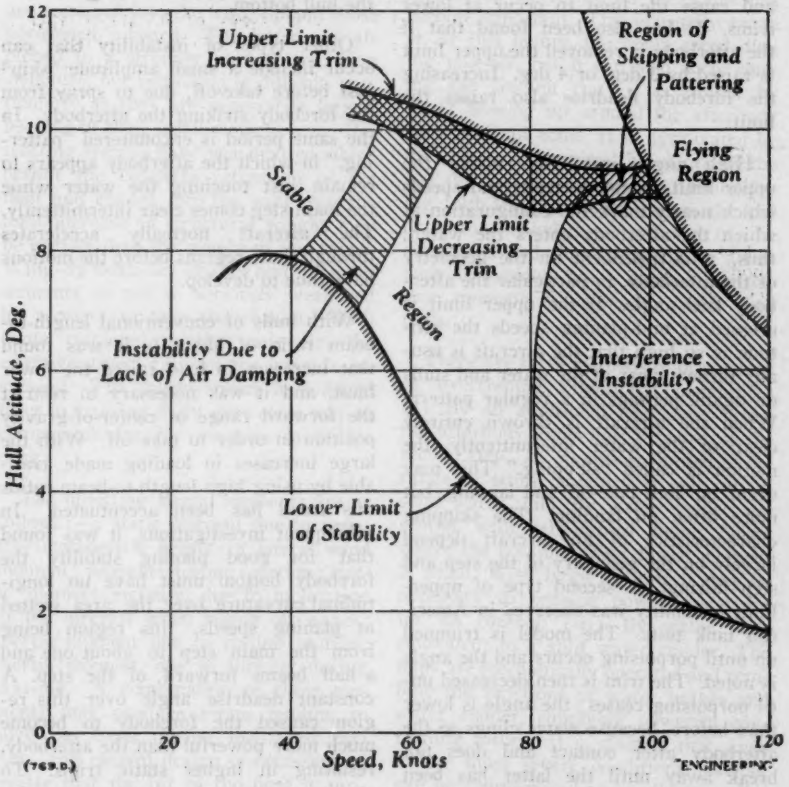
It is a well-known fact that high efficiencies, of the order of a 30-to-1 lift-to-drag ratio, can be obtained with hydrofoils operating in optimum conditions. By the use of such lifting surfaces it should be possible to provide additional water lift at low speeds for low cost in resistance, thus improving the over-all efficiency. Investigations show that a resistance-to-weight ratio of the order of 0.12 should be possible. There are, however, many difficulties to be overcome before the application of hydrofoils can be successful. An excessive rise in suction over the back of the foil, due to speed, attitude or loading, results in cavitation. Furthermore, breakdown of efficient lift may be precipitated if the foil is too close to the water surface. It is difficult, therefore, to ensure stable running conditions. The most successful experiments have had two set of foils, one at the bow and the other—the main lifting foils—just aft of the center of gravity. "Ladder" systems which may include pronounced dihedral have also been used. A possible solution is to use hydrofoil lift up to medium speeds, and, after cavitation to use the planing lift from the lower surfaces. Subsequently, the foils would retract completely into the streamline body.

Hydro-skis obtain their lift from the water pressure on their lower surface in a similar manner to the planing lift of the boat planing bottom, but a considerably higher loading can be used, and consequently a higher attitude of the planing surfaces is necessary. Researches in England and America on the application of hydro-skis to high-speed designs have shown that it is possible to have a highly stable hydro-ski system which is capable of lifting an aircraft body off the water at relatively low speed. The hydrodynamic efficiency of a hydro-ski is low and it is unlikely that a resistance-to-weight ratio lower than 0.35 can be obtained. However, because of their capability of total retraction, hydro-skis are very suitable for water-based fighters where the power available is more than adequate to ensure a short take-off run.

Perhaps the most successful method of estimating spray height is that based on the results of tests, made in America, on a series of hulls having varying length-to-beam ratios and deadrise angles. From these and other results curves have been produced, from which the propeller tip heights can be determined.

Local curvature at the chine, which terminates horizontally, or nearly so, has practically no effect on spray height. Reflex curvature at the chine is found to increase the spray height. Transverse bottom sections which are hollowed from keel to chine are an improvement over straight bottom sections. Spray strips, or plates, which project horizontally from the chine have no effect on spray height and setting the plates at small negative angles to the horizontal shows no benefit, but plates at about 90 deg. have proved successful. Spray strips, parallel to the keel but at varying dis-

Fig.3. WATER STABILITY DIAGRAM.



tances from the chine, are less effective, the reduction in efficiency being proportional to their distance from the chine.

Most flying boats in the past have shown some degree of fore-and-aft instability on the water. With the advent of high loadings and increases in take-off speeds, any form of instability can become uncontrollable. Porpoising, a cyclic oscillation in pitch and heave, is the most dangerous type of instability, especially when it occurs at low trim angles. Analysis of records made in various tanks indicates that trim and heave are out of phase, with trim usually leading.

Two types of porpoising occur, as shown in Fig. 3, which is a plot of hull datum attitude against speed. "Low angle" porpoising, which occurs at trim angles below the lower limit of stability, is due to the hull forebody design, except in the hump region. Tests of a forebody alone show that the lower stability limit occurs at very high trims at low speeds, the trim decreasing as speed increases. The effect of the afterbody is

to damp the motions in the hump region and cause the limit to occur at lower trims. It has also been found that if the afterbody is removed the upper limit is raised by 3 deg. or 4 deg. Increasing the forebody deadrise also raises the limit.

High angle instability, above the upper limit, occurs at trims and speeds which nearly mark the configuration at which the afterbody enters the water; thus, it is dependent on the geometry of the afterbody, in particular the afterbody keel angle. If the upper limit is crossed at mid-planing speeds the attitude is so high that the aircraft is usually thrown clear of the water and stalls on again, perhaps in a regular pattern. When the aircraft is thrown entirely clear of the water intermittently, the motion is called "skipping." This may occur on both take-off and landing, but more often on landing. The skipping characteristics of an aircraft depend largely on the geometry of the step and step fairing. A second type of upper-limit instability was observed in American tank tests. The model is trimmed up until porpoising occurs and the angle is noted. The trim is then decreased until porpoising ceases; the angle is lower than before, because water clings to the afterbody after contact and does not break away until the latter has been pulled appreciably clear of the undisturbed trough. The two limits are called upper limit increasing and decreasing trim, respectively (Fig. 3).

If a craft meets a disturbance in the hump region between the upper and lower limits, instability may occur, due to inadequate tail damping at these speeds. "Interference instability" is experienced with some aircraft at high speeds, where the afterbody design does not allow free entry of air between the bottom and the trough. High suctions then occur, because the limited air supply is entrained and drawn away in the water faster than it can be replaced, so that the water intermittently sticks to the hull bottom.

Other types of instability that can occur include a small amplitude "skip" just before take-off, due to spray from the forebody striking the afterbody. In the same period is encountered "pattering," in which the afterbody appears to remain just touching the water while the main step comes clear intermittently. The aircraft normally accelerates through these regions before the motions have time to develop.

With hulls of conventional length-to-beam ratio of about 6, it was found that increases in load raised the lower limit, and it was necessary to restrict the forward range of center-of-gravity position in order to take off. With the large increases in loading made available by using high length-to-beam ratios this trend has been accentuated. In subsequent investigations, it was found that for good planing stability the forebody bottom must have no longitudinal curvature over the area wetted at planing speeds, this region being from the main step to about one and a half beams forward of the step. A constant deadrise angle over this region caused the forebody to become much more powerful than the afterbody, resulting in higher static trims. To counteract this, the deadrise was increased in a uniform manner, while still maintaining straight buttock lines. It was also found that increasing the forebody warp—i.e., the increase in deadrise angle per beam—improved the lower limit. Increasing the afterbody angle up to about 9 deg. raised the upper limit of stability.

Flying-boat hulls are laterally unstable because their metacenter is below the center of gravity. Many successful flying boats have been stabilized with stub wings, or sponsons, but the view in the United Kingdom is that their extra drag and weight are too great.

Floats can be of two types—those which are retracted into the wing tip and those positioned inboard. The latter are attached to a stronger part of the wing structure, but retraction is difficult, particularly as they are necessarily larger, because of their small moment arm. Retracting systems have been devised which permit the complete retraction of the float and chassis into the wing, but the total increase in structural weight may be greater than the weight of fuel saved. Wing-tip floats are smaller and the wing structure is not so seriously weakened by recesses to take the chassis. With floats retracted to the wing tip, the effective aspect ratio of the wing is increased. The improvement in induced drag in cruising flight goes some way to compensate for the increase in profile drag.

The upsetting lateral moments are as follows:—a static moment due to negative metacentric height; a lateral wind moment; propeller-torque reaction; static moment increase, due to inclination of the aircraft when riding on waves; inertia moment due to rolling motion; inertia moment, due to turning during taxi-ing. A combination of these moments may produce forces tending to bury the floats during low-speed taxi runs, and careful design is necessary to ensure that the lift of the float is progressive with increasing speed. This can only be obtained by float development in the towing tank. It is possible to use a step form which is favorable to low drag increase, since its fore-and-aft stability is unimportant. The float-supporting structure must withstand landing impact in circumstances of dropping wing and adverse cross wind, in addition to the usual taxi-ing conditions. This calls for a high degree of sturdiness. Some flexibility is an advantage, and a cantilever structure is suitable.

Recently, due to changed configurations and heavy loadings, the problem of directional instability on the water has become important. This is usually worse at, or just before, the hump speed during take-off, but it can also occur during the landing run; it is caused by water flowing up around the afterbody sides and tail cone. It is aggravated by the sides overhanging the chines of the last third of the afterbody. By employing high-length-to-beam ratios, the tail extension behind the second step can be eliminated and thus overhanging afterbody is avoided, but a greater controlling moment is necessary with long hulls.

There are three methods of preventing or overcoming directional instability: (i) By an additional keel plate below the afterbody extending forward of the second step. Its success is limited to the range of trims at which it will run in solid water. (ii) By water rudder, essential with jet-engined flying boats, where the possible offset thrust moment is small. With large aircraft it may be necessary to provide power assistance for its control. (iii) By water flaps equivalent to a split water rudder. The flaps are mechanically operated and combine the functions of water brakes and water rudder, supplying powerful controlling forces. It is claimed that, even in the most severe conditions, yawing can be counteracted and a directionally-stable take off can be made.

The sea in which the flying boat is to alight will most probably have a major swell interfered with by a lesser swell from another direction, on which are superimposed waves running in the direction of the local wind. The length-to-height ratio of waves varies from over 100 to 1 down to 7 to 1; below this the waves are unstable and break. Long swells exceeding, say 750 ft. and of moderate height are not dangerous provided that the wind component along the swell is less than 20 knots, for then the flying boat can run parallel to the wave crests. If the wind exceeds 20

knots, operation into wind is probably essential because of directional stability, drift and reserve buoyancy considerations. In contrast, severe impact loads and bouncing can occur in relatively short steep seas. A long shallow swell can also produce high accelerations if the craft is thrown off the crest. The most critical wavelength for any particular hull appears to be between two and three times the hull length. If the sea has a predominant heavy swell in a given direction, it may be preferable to alight along the swell even when the local wind is on the beam. With suitable hull design it has been found possible to alight safely in seas up to 9 ft. from trough to crest.

In the United Kingdom theoretical estimates of peak accelerations have given values agreeing well with full-scale results obtained in calm and rough waters. This work has made it possible to approach, with confidence, an assessment of the impact loads necessary for the structural design of future flying boats. Calm water landings of a Sunderland V, which recorded impact accelerations of 1.0 to 1.3 *g*. gave mean pressures between 12 and 15 lb. per square inch, with peak pressures of about 25 lb. per square inch recorded locally. These pressures are of very short duration, of the order of ⅛ second. Tests on a Sikorsky S.43, showing a similar acceleration (1.3 *g*), recorded average bottom pressures of about 9 to 10 lb. per square inch, with peak pressures which vary from 15 lb. per square inch at the keel to 18 lb. per square inch at the chine. These values are representative of normal landing cases, but in rough seas, particularly with heavy swell, much higher accelerations have been recorded. In some notable tests recently made on a Mariner and Mars, where landings were made into exceptionally heavy seas, acceleration as high as 5 *g* were measured. At the same time, rotational angular accelerations were recorded, sufficient to produce accompanying acceleration at the bow or tail of 9 *g* and, in one case, 11 *g*. It is noteworthy that the structures of the two aircraft concerned were capable of withstanding the abnormal loads. It is unusual, in normal practice, for accelerations to be developed which exceed 2.5 *g* at the center of gravity.

Statistics of structure-weight percentages covering the past ten years have shown that the upward trend, predicted by the square/cube law, has not so far been realized; in fact, the largest aircraft, in the 300,000 lb. range, show a lower percentage than the average medium-sized aircraft. Factors contributing to this are as follows. (i) The increase of wing loading with size: the maximum limit in wing loading has not yet been reached, and this assistance to weight alleviation will continue for further size increases.

(ii) Materials of higher specific strength can be employed advantageously in large aircraft, where it pays to design each part exactly to suit the load, whereas the size of many parts on small aircraft is decided by the necessity for handling or general sturdiness.

(iii) In the past, lower strength factors have been found to be permissible with large aircraft since, owing to the slower acceptable rate of control response, the accelerations to be met in flight are less. Recent gust requirements, however, show a tendency to increase and new ideas on the cumulative effect of repeated loading may result in increases in structural weight.

(iv) By more careful consideration being given to the disposition of the main structural members on large aircraft, it has been found possible to use material more economically.

(v) A number of items included in the structure weight are common to all sizes of aircraft.

In modern designs of flying boats, the ratio of profile drag to all-up weight shows a progressive improvement as the size increases. This can be ascribed to the trend towards higher loading as size increases; the proportionately smaller body surface area required to contain the payload; more easily controlled parasitic and interference drag; and, in the larger aircraft, the power plant can be buried. There is, therefore, much to be gained by increasing the size of aircraft.

Flying boats should remain in the water for as long as possible, and all short-period maintenance work, refueling, etc., should be done afloat. The floating pontoon dock appears to be the best solution to this problem, preferably U-shaped so that there is access to both sides of the boat. If the dock is connected to the shore it is relatively simple to run refueling and electrical-supply lines to points on the dock. All loading and unloading can be done at the dock and facilities can be provided for maintenance access. All major overhaul work should, however, be done out of the water, at the maintenance base, which should have a slipway, hard standing, hangars and stores. It will also have a number of permanent pontoon docks with all necessary supplies piped from the shore. Emergency bases, which could only be used infrequently, need not be equipped with more than fuel storage and means of supply.

It is quite usual for flying boats to be landed at sea in emergency or for rescue purposes. By special design, regular landing in the open sea can be included in the aircraft's military duties, thus extending considerably the localities in which the aircraft can be operated.

Capability of long range is one of the flying boat's most valuable assets. Future long-range aircraft will almost certainly use propellers. The choice lies between propeller turbines, piston engines and compound engines. The propeller turbine is suitable for the transport aircraft where high-speed operation at high altitudes is permissible. This type of engine, however, lacks flexibility in operation and a strict flight plan must be followed, using maximum cruising power at the maximum altitude throughout the flight. The piston engine is obsolescent for long-range aircraft, although it still has a future for feeder-line aircraft or amphibians. The newest development in the compound engine is exemplified in the Napier "Nomad," which has an exceptionally low specific consumption. Its flexibility makes it particularly suitable for ocean-patrol flying boats which are required to operate over a wide range of altitude and speed.

The transport of troops by air, probably in converted civil air liners, will become universal in the future, not only because of the advantage in rapid reinforcement but in order to eliminate the loss of effectiveness while troops are on long journeys by sea.

Aircraft designed for passenger-carrying are, however, incapable of accommodating the equipment and supplies which must be available to the airborne troops. The transport flying boat appears to be the answer to this problem, since it is virtually unrestricted in size and does not require prepared aerodromes. Placing the main doors in the bows simplifies loading and makes it possible to use the flying boat as an assault craft, which can be run, bow first, on to a suitable beach. Flying boats of 330,000 lb. total weight are soon to fly in Great Britain, and in America a boat of 450,000 lb. has been built. The limiting factor in size in the future will probably be the availability of engine power and the difficulty of converting high shaft power efficiently into thrust. For this reason the author would set a limit to the upward rise in weight at about 600,000 lb., for some time to come.

The qualities the passenger will expect in order to attract him to a par-

ticular airline are considered to be a smooth ride, short duration journey, quietness, and comfort. The smooth ride means operating at heights above 20,000 to 25,000 ft. A short duration of flight calls for both a high cruising speed and long range, two attributes difficult to attain simultaneously. Without sufficient range, refueling stops must be made, which greatly reduce the average speed on the over-all journey. It is perhaps easier to obtain a quiet aircraft with propeller-turbine power plants than with other types of power; by the use of a large aircraft, the distances between sources of noise and the cabin walls can be increased, with beneficial results. The amount of comfort provided should increase with the duration of the trip. It has been found possible to meet many of these desired qualities in designing the Princess, by reason of its size and the use of propeller turbines, which show their most economical operation at high altitude and at relatively high speed.

For medium ranges the jet aircraft can be a better economic proposition than a propeller-driven aircraft, especially if a return journey can be made in one day on a route where the lower-speed aircraft can only schedule a oneway trip. It is thought that the jet-engined passenger aircraft will eventually become universal for stage distances of 1000 to 2000 miles. Fig. 4 shows a flying-boat project designed to operate on stage distances of about 1500 miles at a cruising speed of 450 m.p.h. The maximum take-off weight is 130,000 lb. and the aircraft is fitted with six de Havilland Ghost engines. There does not appear to be a great gain in operating a passenger aircraft with a higher cruising speed than 500 m.p.h., and it is probable that future development will be in range and size.

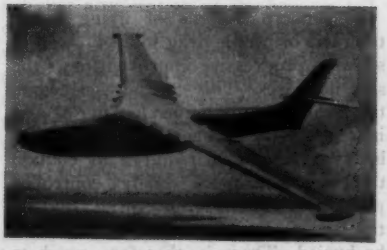


FIG. 4. JET-PROPELLED AIR-LINER DESIGN.

The operation of intruder-bomber flying-boat squadrons from remote parts of the world is being considered. By using improvised bases, the radius of action can be reduced to distances which permit the design to be primarily for speed. This form of bomber can also be used for minelaying.

If a water-based fighter is to compete on equal terms with a land-based fighter, no concession can be made during flight to hydrodynamic requirements. The conventional flying-boat hull must be discarded. Fortunately, the power necessary for high performance in flight makes it possible to adopt low-efficiency hydrodynamic systems for water operation, and to concentrate on a blending of the characteristics which contribute to both high-speed flight and satisfactory water behavior. Fig. 5 shows the Convair solution for an attack fighter-bomber. Messrs. Saunders-Roe, Limited, have designed a high-speed water-based fighter, shown in Fig. 6, above, in which hydro-skis are used. Although capable of taking off from water, it is no longer a flying boat in form. In this design the jet engine has its air intake above the fuselage in a form of Pitot entry of high over-all efficiency. These water-based fighters illustrate the future possibilities of aircraft which are independent of runway

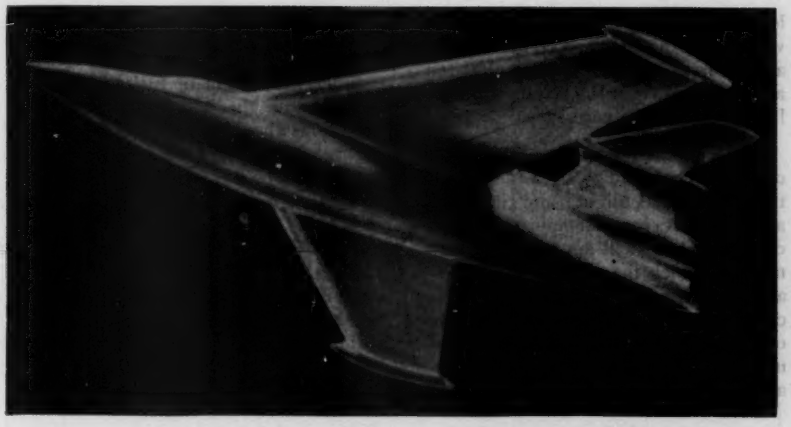


FIG. 5. CONVAIR" WATER-BASED FIGHTER-BOMBER DESIGN.

construction and, therefore, are capable of use in any area of the world, provided that there is sheltered water from which to operate.

It has recently been announced in the United States that work has been initiated on an aircraft deriving its power from nuclear fission. Its initial application might well be associated with a flying boat, for the following reasons. It is probably safe to assume that only the heat energy of nuclear reaction can be used and that vapor-driven turbine engines with propellers will form the power units in early installations. The weight of reactor unit, controls, and screening will be high. This part of the installation takes the place of the fuel load in the conventional aircraft. A very approximate approach to the aircraft weight indicates that for structure, engines, take-off assistance and useful

FIG. 6. SAUNDERS-ROE WATER-BASED FIGHTER DESIGN.

load, roughly 60 percent of the total weight will be required. Therefore, for a reactor unit weighing, say, 100 tons, an aircraft of from 500,000 lb. to 600,000 lb. weight will be necessary.

The atomic-powered aircraft will cruise at constant weight and power throughout the flight, and it can be assumed that, for economic reasons, the cruising power will approach the maximum of which the plant is capable. As a consequence, for take-off and initial climb, assistance from some other form of energy, such as liquid rockets, will be necessary. In order to reduce power assistance as much as possible it will be wise to make the take-off from water, where runway length is unrestricted.

Since the weight of fuel consumed is insignificant, the optimum aerodynamic conditions should prevail throughout the flight, and the take-off and landing weight will be the same. The possibility of landing flying boats at their take-off weight has always been recognized as an acceptable maneuver. The unobstructed approach run, unrestricted landing area, and short run after touch-down reduce the landing hazard, and the shock-absorbing properties of a V-bottom ensure low landing accelerations even at overload weights.

THE NATIONAL SCIENCE FOUNDATION IN RELATION TO BASIC ENGINEERING RESEARCH

DR. ALAN T. WATERMAN

THE AUTHOR

Was born in Cornwall-on-Hudson, New York, in 1892. Received A.B., A.M., and Ph.D. degrees from Princeton University. He was a physics instructor at Cincinnati University in 1916-17 and at Yale University from 1919 to 1922. From 1923 to 1930 he was Assistant Professor of Physics at Yale, and in 1931 became Associate Professor. In 1927-28 he was a National Research Fellow in Physics at King's College in London. In 1942 he became Vice-Chairman, Division D, National Research Defense Committee. From 1943 to 1945 he was Deputy Chief, Office of Field Service, Office of Scientific Research and Development, and in 1945 became Chief of this Office. He became Chief Scientist, Office of Naval Research, Navy Department in 1946, which position he held until he was nominated to be the first Director of the new National Science Foundation in 1951. He has been Chief Reader and Chief Examiner for physics, College Entrance Examination Board, since 1931. He is a fellow in A.A.A.S., The American Physical Society and the American Association of Physics Teachers; and a member of the American Association of University Professors, Phi Beta Kappa, and Sigma Xi. As an author and as an editor he has made important scientific contributions to many technical journals. Dr. Waterman served in World War I as an officer in the Army Signal Corps.

After some five years of legislative debate in Congress, the National Science Foundation was brought into being by the National Science Foundation Act (Public Law 507—81st Congress) of May 10, 1950. The creation of this new agency marked a significant step by the Government in recognizing the importance of science to our national life. It marked the establishment of an independent agency at the policy-making level of government to insure continuing support and emphasis to basic research and education in the sciences. Although a number of federal agencies have sustained various types of research and development programs, some of which go back to a very early date in American history, there had never been a single agency charged with the specific responsibility of developing and encouraging the pursuit of a national science policy toward the furtherance of progress in science itself.

The Department of Defense, and particularly the Office of Naval Research, may be proud of its record for supporting basic research in the sciences

through contracts with universities and other research institutions. The ONR program began in 1946. At the same time, other government research groups, such as those in the Department of Agriculture, were supporting certain scientific investigations. Normally, these research programs tend to emphasize research in fields related to the missions and objectives of the sponsoring organizations. These missions and objectives, therefore, are the limitations of each program, however large that program may be. In addition to encouraging such basic research programs, the Foundation has as its mission the growth and development of American science.

An examination of the functions of The National Science Foundation, as enumerated by the Foundation Act, will indicate the number of ways in which its mission is unique.

FUNCTIONS OF THE FOUNDATION

The Foundation is authorized and directed—

(1) to develop and encourage the pursuit of a national policy for the promotion of basic research and education in the sciences.

(2) to initiate and support basic research in the mathematical, physical, medical, biological, and engineering sciences and to appraise the impact of research upon industrial development and the general welfare.

(3) to initiate and support at the request of the Secretary of Defense specific research activities.

(4) to award scholarships and graduate fellowships in the mathematical, physical, medical, biological, and engineering sciences.

(5) to foster the interchange of scientific information.

(6) to evaluate scientific research undertaken by agencies of the Federal Government and to correlate the Foundation's research programs with other such programs.

(7) to maintain a register of scientific and technical personnel.

(8) to cooperate in international scientific research activities.

To fulfill these broad functions, four divisions were established within the National Science Foundation: a Division of Medical Research; a Division of Mathematical, Physical and Engineering Sciences; a Division of Biological Sciences; and a Division of Scientific Personnel and Education. Bolstering the work of the research divisions are Divisional Committees appointed by the National Science Board, and Advisory Panels and Special Commissions appointed by the Director.

In its program of research support, the Foundation will make use of existing facilities and, where appropriate, of experienced teams of investigators. It has no power to establish or operate laboratories of its own. The Foundation is concerned with the stimulation and encouragement of basic research as contrasted with applied research which, for the most part, enjoys abundant support.

Heretofore two methods have been widely used by government agencies for financing research. One of these is the fixed-cost contract which requires that the cost of the work be estimated in advance in rather detailed form. Furthermore, it calls for a specified performance for an agreed-upon compensation. The other method is the reimbursement-type contract which provides for compensation for all costs incurred in the accomplishment of the work. The latter type has been more widely used since World War II. Both methods require rather elaborate bookkeeping, cost-accounting, etc., on the part of the contract administrators and the research scientist or his institution.

The National Science Foundation Act provides a large degree of freedom in the choice of means for supporting research. In most cases, such support is given through direct grants for the conduct of research. The typical grant instrument is exceedingly simple. Under it the research worker is not overburdened with bookkeeping. He agrees to carry on his research under a statement of aim and scope of the work. His performance is largely the result of his ability and integrity in carrying out his intentions.

Since the Foundation does not plan to support programmatic research—that is research designed to answer specific problems, and since it judges proposals largely on the ability of the investigator and the scientific significance of the project, the grant seems particularly appropriate for the support of basic research by the Foundation.

In the practice of both private and public institutions, a grant is a donation of money, property, or other valuable thing usually for the accomplishment of an agreed-upon purpose. Like the contract, the grant takes different forms depending upon the purpose to be achieved and the relationships established between parties. Typically, the grant is characterized by statements of objectives and essential undertakings. Emphasis is placed on informal understandings, although legal protection against deception and noncompliance is preserved. It is anticipated that most National Science Foundation grants will be awarded to institutions for the work of individuals of demonstrated competence. It has been found that investigators tend to submit proposals because of their keen interest in exploring some particular scientific area. At the same time they lack the means necessary to carry on the work with expectation of reasonably early results.

The principal fields to which support will be given are, as implied by the structure of the Foundation: medical research, the biological sciences and the mathematical, physical and engineering sciences. The program in the latter fields is of chief interest to engineers.

THE DIVISION OF MATHEMATICAL, PHYSICAL AND ENGINEERING SCIENCES

The Division of Mathematical, Physical and Engineering Sciences is the largest in scope of the four divisions. It is headed by Dr. Paul E. Klopsteg, who has been granted a leave of absence as Professor of Applied Science and Director of Research of the Northwestern Institute of Technology. Dr. Klopsteg has been with the National Science Foundation some four months, a relatively short time to get a research program underway. The operation of the division, therefore, is still in the formative stages. Key personnel are still being recruited for the program. In view of the enormous scope of the subject matter included in this division, the work has been subdivided into five categories: (1) mathematics and astronomy; (2) physics and theoretical mechanics; (3) chemistry; (4) earth sciences; and (5) engineering and metallurgy.

The fact that the National Science Foundation is directed to support basic research in engineering, which is specifically labeled as a science along with the mathematical, physical, medical, and biological sciences, creates what appears at first glance to be a paradox. The world of science and technology has long been accustomed to think of basic research as one step in the process of research and development, and engineering as a much later step. Gano Dunn has, in fact, described engineer-

ing as "the art of the economic application of science to social purposes."

The idea of supporting basic research in engineering may not seem so paradoxical if one makes a new approach to the concept. A recent address by Dr. Klopsteg to the Engineering College Research Council takes this new tack. He said:

> Ideally, the scientist has but one aim: to penetrate and conquer the areas of ignorance and to annex the territory thus gained to the contiguous areas of knowledge. Perhaps this description may serve as a working definition for basic research; perhaps also it may help our thinking about basic research in engineering. If penetration and conquest of ignorance is an essential step in research, engineering may lay equal claim, with science, to the procedure; for science has no monopoly of that which is unknown. It seems quite unprofitable to try to write specifications for ignorance, or to subdivide it into areas. The point is that lack of knowledge about applications of scientific principles to a particular situation is ignorance just as truly as is lack of knowledge of the principles to be applied. Hence it seems logical to take the position that research directed towards methods and procedures of applying knowledge towards useful ends is research as truly as that which seeks knowledge of science in the first instance. I am mindful, furthermore, that in seeking to devise procedures and methods of application, engineering research must always be concerned with the economic and social aspects of its possible results. This makes the research no less basic.

These are the broad ideas which underlie the plans for the National Science Foundation program for supporting basic research in engineering. Proposals for study in this area, as in other scientific areas supported by the Foundation, originate with the investigator. He states the problem, outlines the method of attack, and suggests the scientific implications. In the evaluation procedure much attention is paid to the competence of the investigator.

Actually, in carrying on the work of The National Science Foundation, there are no strict rules or standards of measurement which can be applied to a specific proposal to determine whether it is basic research. Each situation will be evaluated with the best judgment which can be brought to bear on it, and decisions reached in harmony with the broad objectives of the Foundation. It is the intent that basic research in engineering be supported, whether or not the subjects to be investigated bear the specific label "engineering."

Some examples of the subjects in engineering in which basic research may lead to significant knowledge include:

Transient flow in porous media.

Fatigue characteristics of engineering materials under conditions of variable cycle loading.

Fluid friction in unsteady motion.

Surface temperature of sliding surfaces.

Theoretical study of detached shock waves.

Study of the earth as a thermal source and sink.

Solar energy studies.

Diffusion in liquids near the critical point.

Influence of ultrasonics and microwaves on chemical reaction rates.

Theoretical studies of interaction phenomena of electric charges and electromagnetic fields.

Properties of wave guides, wave guide filters.

Power measurement at radio and microwave frequencies.

New methods of performing mathematical operations with electronic computers.

Intermetallic compounds as high temperature materials.

Correlation of crystal structure of refractories with thermal shock properties.

Correlation of dielectric properties of mica with crystal structures.

Plastic flow and stresses within the grain of metals.

Oxidation of metals at high temperatures and pressures.

Application of measurement and control to research problems in various sciences.

THE DIVISION OF SCIENTIFIC PERSONNEL AND EDUCATION

The current shortage in engineering manpower is, of course, well known. This is a principal concern of the Division of Scientific Personnel and Education, headed by Dr. Harry C. Kelly. Dr. Kelly was formerly Chief Scientist of the Office of Naval Research Chicago Office, and prior to that Chief of the Scientific and Technical Division which advised General MacArthur on the development of a scientific and technical program for Japan.

The Division is responsible for the development and administration of programs relative to the granting of graduate fellowships in the various sciences. The Foundation hopes, by means of its graduate fellowship program, to be able to augment, at as early a date as possible, the supply of scientific manpower. As of January 1951, some three thousand fellowship applications were received, from which five hundred fellowships will be awarded for the 1952-53 academic year. It is hoped that, for the following academic year, this program can be expanded considerably.

The degree to which the defense program may be limited by insufficient numbers of trained scientists and engineers is a matter of national concern. In August, 1951, Charles E. Wilson, Director of Defense Mobilization, issued a strong statement calling attention to the critical nature of the problem. After stating that the supply of scientific and engineering graduates in 1951 was less than half that needed to man America's economic and defense program adequately, Mr. Wilson suggested three important ways in which the shortage could be remedied:

(1) Government, industry and educational institutions, must make the most effective possible use of those persons who have scientific and technical training.

(2) Educational institutions should develop counseling programs to train a larger number of men and women in these fields.

(3) Industry and government should develop on-the-job training programs and cooperative training programs with educational institutions through which employed persons will receive scientific and technical training.

The National Science Foundation has already started the necessary surveys and analyses to provide a basis for a sound and realistic national policy for science and science education. At the same time, the fellowship program in the mathematical, physical, medical, biological, and engineering sciences will in part relieve the present manpower shortage.

METAL ECONOMICS SCRAP RECLAMATION, SECONDARY METALS, AND SUBSTITUTE METALS

The Scope for Conservation of Metals, Ferrous and Non-Ferrous

BY C. A. BRISTOW, B.SC., A.R.S.M., A. J. SIDERY, ASSOC. MET., and H. SUTTON, D.SC., OF THE MINISTRY OF SUPPLY

ACKNOWLEDGMENT

This is Part II of an article titled "Metal Economics" which was published in the January 1952 issue of the "Journal of the Institute of Metals." Part I was reprinted in the May 1952 issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

Metal economy can usefully start in the design stage. It is specially important that each particular product be considered critically in such ways as the following:

(*a*) Can the required performance be secured with the use of materials that are less difficult to obtain?

(*b*) Does the design include anywhere material which could be economically excluded as being in excess of what is really required?

(*c*) Can a less wasteful method of production be employed, without or with practicable modifications of design?

(*d*) If good protection against corrosion or better resistance to creep were reasonably practicable, could less metal be used to produce a satisfactory or superior article?

FERROUS MATERIALS

The shortage of alloy additions to steel emphasizes the importance of the use, wherever possible, of low-alloy steels. The situation regarding alloy additions in steel differs somewhat from that prevailing during the last world war, but the experience gained then should assist greatly. The compositions of these steels in the light of information on current and prospective availability of the different ferro-alloys and additions, and the properties attainable by suitable and special techniques of steel-making, working, and heat-treatments in relation to sections of different mass will undoubtedly receive much study by manufacturers and users. Redesign and intensive testing are involved for many applications.

Low-tungsten high-speed steels or cutting materials free from tungsten, if capable of extensive use, would greatly relieve the difficult tungsten position.

In connection with stainless and heat-resistant steels, recent important developments will contribute substantially towards economy in the use of niobium, except in cases where welding or exposure to high temperature, or both, are concerned. The practicability and efficacy of low carbon content, and the value of titanium as a stabilizer, are well recognized by metallurgists, but possibly not yet sufficiently appreciated by engineers. In view of the special properties of niobium and of the very limited amount available, it is important that it should be used only in essential applications, such as in heat-resisting steels and alloys and welding materials.

In the gas-turbine field substantial economies have already been achieved in alloying metals by the use of ferritic materials for discs—a development made possible by engineering design. Further economies may be realizable, but for special applications, such as gas turbines and steam plant, strong claims for the special elements used in heat-resistant materials will undoubtedly call for renunciation in other fields.

NON-FERROUS MATERIALS

Copper, zinc, aluminum, and lead are the non-ferrous materials produced in greatest quantity, and for each of them, with the exception possibly of aluminum, demand at present exceeds supply. Aluminum has already found applications which, if it were not available, would require larger amounts of other materials for duties which, in many cases, are performed more effectively by aluminum.

Large amounts of copper are used in condensers for power plants. If aluminum or aluminum alloys can be used here, without corrosion problems for the user, much copper can be applied to other uses. The use of aluminum instead of copper for bus bars would seem to be obvious.

In the cable and conductor field the increasing use of synthetic materials, such as polyethylene, protected by a light sheathing of aluminum instead of lead, is a matter of necessity. Battery plates made in lead alloys having improved mechanical properties appear likely to save lead, though performance in service is an important part of these developments.

An interesting development is the introduction of organic adhesives for joining metals to replace riveting and welding. Reinforced plastics, impregnated timber, etc., as structural materials are undergoing intensive study. Further economies in the use of valuable metals can be effected by the wider application of reinforced plastics as bearing and

gear materials. The successful application of plastics to replace metals calls, however, for very careful consideration by the designer.

SCRAP SEGREGATION AND RECOVERY

Increasing efficiency in the use of scrap and used material must be achieved. Miller forecasts that, unless an expansion of the production of low-grade alloys from scrap is achieved, it will be necessary to consider the adoption of processes for the treatment of scrap to recover the constituent metals in a high state of purity.

During the last world war, and to some extent since, scrap segregation, making possible efficient re-use of the material, proved of great advantage. Color markings for scrap from particular groups of light alloys or steels permitted the recovery of large amounts of valuable material. There is scope for much more action on these lines, thereby saving labor as well as material. In connection with the identification of alloys for secondary handling, a great deal of information and experience is available, ranging from rapid tests made in simple ways to the more elaborate analytical methods, including spectrography. Fewer different alloy specifications would help in this direction.

CORROSION LOSSES

Uhlig has estimated the value of the direct annual loss by corrosion and in combating corrosion in U.S.A. to be of the order of 5500 million dollars, and J. C. Hudson puts the loss in connection with corrosion and protection of iron and steel in the United Kingdom at £200 million annually. In many cases the engineer has to face risks of great consequential expenditure that may arise from a failure by corrosion and tends to over-design his pipes or other equipment to reduce that risk. Renewals made necessary by corrosion are not the only serious aspect of the subject, but these alone account for substantial demands for the metals involved. Great economies have been achieved in particular fields by the development and use of corrosion-resistant materials and of improved methods of protection of industrial metals. These aspects of protective coatings for metals have recently been presented by U. R. Evans.

THE FORWARD VIEW

Few indeed are the industrial metals for which there is encouragement to hope for greatly or rapidly increased rates of world production. Of the elements present in substantial amount in the earth's crust, silicon has hitherto defied the attempts which have been made to render it ductile. Ductile, workable silicon or silicon-rich alloys might do much to relieve present and prospective demands for industrial metals and alloys.

Aluminum is nearly a thousand times as abundant as copper in the earth's crust, and is one of the metals likely in future years to be called upon to serve mankind to a much greater extent than at present.

Magnesium is nearly 250 times as abundant as copper, and the raw material which is the main source of the metal in the U.S.A. and the United Kingdom is sea-water. Sea-water contains about 1272 mg. of magnesium per kilogram, and Teed mentions that a cubic mile of ocean water contains about 5,400,000 tons of magnesium. Present world production is on a much smaller

scale than for aluminum, but magnesium could be manufactured in much greater amounts than at present. Modern magnesium alloys have working properties, mechanical properties, and durability such as to make them capable of much more extensive use in general engineering and many other applications. Alloys of high resistance to corrosion are now available in cast and wrought forms.

Titanium is about 65 times as abundant as copper in the earth's crust. Its oxide is an article of commerce, the amount now produced per day in the pigment industry in America being equivalent to 300 tons of metallic titanium content. The metal itself possesses very good corrosion-resistance and general mechanical properties. Titanium alloys of various types are known which offer good corrosion-resistance and general mechanical properties comparable with those of high-tensile steels. The low density (4.5-4.6 g./c.c.) of these materials, coupled with their general properties, claims for them a considerable future as materials of construction. Production of titanium is at present on a small scale, but is being advanced rapidly in the U.S.A. Important research and development work is in progress in the U.S.A., the United Kingdom, and other countries.

The authors thank the Chief Scientist, Ministry of Supply, for permission to present this paper.

Economy by Standardization of Alloys and of the Method of Reclamation of Scrap Metals

BY C. DINSDALE, M.SC., RAILWAY EXECUTIVE, EASTERN AND NORTH EASTERN REGIONS

British Railways are substantial users of many kinds of ferrous and non-ferrous metals and alloys. Up to 1948 the four main-line railway companies were each using metals—especially non-ferrous ones—with widely differing compositions for the same component. Non-ferrous metal reclamation policy also varied. For many reasons it is most desirable, and indeed essential, that in future all Regions shall use, if at all possible, the same metal for the same purpose.

British Railways are at present using between 20,000 and 30,000 tons of non-ferrous metals and alloys each year on their rolling stock alone. The most recent ex-L.N.E. Class A.1 locomotive, for example, contains nearly 5½ tons of non-ferrous metals and alloys: 3¼ tons of copper firebox plates and stays, 38 cwt. of bronze axle-boxes, 3 cwt. of bronze boiler mountings, 2 cwt. of bronze bushes and bearings, and 100 lb. of white metal being the main items. In addition, non-ferrous metals are used by the civil engineer in buildings, by the signal and telegraph engineer for cables and telephones, and by the electrical engineer for transmission purposes and for overhead wiring and bonds on electrified systems.

Nationalization of the British main-line railway companies as a result of the Transport Act, 1947, made it essential that standard non-ferrous metals and practices should exist in all regions, and in 1948 the member of the Railway Executive for Mechanical and Electrical Engineering set up a Committee to investigate the matter. A brief account of

TABLE XV.—*Bronzes and Brasses for Future Use.*

Type of Alloy	Nominal Chemical Composition, %					Main Uses	Notes
	Cu	Sn	Pb	Zn	Others		
Phosphor-bronze	87.5	12	..	..	P 0.15 (min.)	Slide valves (superheated engines); unlined bushes; machinery bushes and details.	B.S. 1400-PB2
Leaded bronze	67.5	5	26	..	Ni 1.5	Slide valves (saturated engines); regulating valves (grid type).	..
Leaded gun-metal	86	7	2	5	..	Steam and boiler mountings; axle-boxes, lined bushes and brasses; carriage and wagon bearings, &c.	B.S. 1400-LG3
Yellow brass	66	..	2	32	..	Name and number plates; carriage fittings, &c.	B.S. 1400-B3
Brazing metal	90	..	..	10	..	Fittings which are to be brazed on to copper or steel pipe.	..

this Committee's main findings and recommendations is as follows:

(a) *Cast Bronze and Brasses.*—The large majority of the bronze and brass castings required for British Railways' rolling stock and machinery are made in the Railway Executive's 17 "brass" foundries, a total of some 11,000 tons of non-ferrous castings being produced annually. In 1948 these foundries between them were casting 37 copper-base alloys, each of which had a different composition. Fourteen mixtures were being used in small quantities only (less than 10 tons of each per year), while the two most important were 85:5:5:5 leaded gunmetal (over 4000 tons, or about 40% of the total) and 13¼% tin bronze (3500 tons, about 35%). In addition, a further alloy was specified for certain purchased wagon bearing shells.

The five alloys listed in Table XV were recommended by the Committee, and have already been introduced in all Railway Executive foundries for the bronze and brass castings which are made for use on rolling stock and machinery.

It will be noticed that the 85:5:5:5 leaded gunmetal has not been included in this new list. Although large quantities of that alloy have been used for many years in locomotive construction, it was considered that a somewhat harder and better-wearing bronze was required, especially for components such as boiler mountings and axle-boxes. It is hoped that the 7% tin leaded gun-metal will give this increased life and so justify its extra cost. Certain axle-box trials are being made in the Eastern and North Eastern Regions using 7% tin leaded gun-metal instead of the 13¼% tin alloy which has been the standard for many years, and also with the high-leaded bronze for slide valves. If the new alloys can be used successfully for the purposes suggested, complete standardization will have been achieved and a list of 38 bronze and brass castings alloys reduced to five.

(b) *White Metals.*—The Railway Executive probably owns between 6000 and 7000 tons of various white-metal alloys. In 1948 some 18 white-metal alloys with 13 different compositions were being used for lining such components as locomotive and tender axle-boxes, bushes and brasses, eccentric straps, crossheads, and carriage and wagon bearing shells. The white metals being used, however, fell into four well-defined groups, viz: (1) high-tin, lead-free, (2) high-tin, low-lead, (3) intermediate or tin-lead-base, and (4) leadbase alloys.

TABLE XVI.—*Chemical Composition of White Metals.*

Type of Alloy	High-Tin, Lead-Free	High-Tin, Low-Lead	Lead-Tin Base	Lead Base
Tin, %	84–86	80–85	58–60	11–13
Antimony, %	9–11	9–11	9–11	12–14
Lead, %	0.2 max	0–5	27–29	73.5–75.5
Copper, %	4–6	4–6	2–4	0–1
Max. total impurities, % (excluding lead)	0.30	0.30	0.35	0.40
Aluminum shall not exceed	0.05	0.05	0.05	0.05
Arsenic shall not exceed	0.10	0.10	0.15	0.20
Bismuth shall not exceed	0.08	0.08	0.08	0.08
Iron shall not exceed	0.08	0.08	0.08	0.08
Zinc shall not exceed	0.05	0.05	0.05	0.05

It was considered that, as a first step towards standardization, the number of compositions used should be reduced to four, one metal being specified to cover each of the four groups. The compositions of the four alloys now standard are those in Table XVI.

A practice which uses only one metal for all bearings is obviously attractive, owing to the simplicity of lining practice, scrap segregation, and storekeeping of ingot metal. The present high price of tin, however, immediately rules out the introduction of the 80% tin metal for all locomotive, carriage, and wagon bearings; and even the universal use of 60% tin white metal would involve the Railway Executive in additional expenditure. At present, therefore, extensive trials are being carried out on all Regions with the use of these lower-tin-content white metals on locomotive, tender, carriage, and wagon bearings.

(c) *Packing Metals.*—Although much cast iron and non-metallic material is used for such purposes as piston-rod and piston-valve-rod packing, seven non-ferrous alloys were also in use.

The three alloys listed in Table XVII have now been introduced for the purposes indicated. The financial savings are, however, not large, as the quantities involved are small.

TABLE XVII.—*Metallic Packing.*

Type of Alloy	Nominal Chemical Composition, %		Typical Use
Antimonial lead	Antimony Lead	20 80	Piston-rod and piston-valve-rod packing; saturated engines.
Copper-lead	Copper Lead Nickel Sulphur	37 61 1.25 0.75	Piston-rod and piston-valve-rod packing; superheated engines.
Leaded-bronze	Copper Lead Tin Nickel	67.5 26 5 1.5	Piston-rod, piston-valve-rod and McNamee packing; superheated engines.

(d) *Wrought Metals.*—There was already considerable agreement among the Regions both as to the composition of the wrought non-ferrous metals being used in the construction of rolling stock and the purposes for which they are used. This was undoubtedly due to the fact that well-known and long-established British Standards existed for these materials, viz. B.S. 24, Part 5, B.S. 218, B.S. 249, etc.

(e) *Solders and Brazing Metals.*—In the main, the composition and uses of soft solders and brazing metals used were based on the appropriate British Standard, e.g., 219 (soft solders), 263 (brazing spelter), and 206 (silver solder), but in a few instances solders with other compositions were being made and used. Small amounts of proprietary brazing materials were being used. Some 70 tons of soft solders are made each year in Railway Executive shops and 20 tons purchased. About 15 tons of brazing solder and ½ cwt. of silver solder are also bought.

As British Standards exist, or will shortly be issued, for most grades of soft and silver solders and brazing metals, the Railway Executive have not found it necessary to issue many of their own specifications for this class of material.

(f) *Aluminum and Magnesium Alloys.*—Although for many years certain Continental and American systems have made considerable use of light alloys in rolling-stock construction, in this country comparatively little use is being made of such alloys. "Alpax" vacuum and steam-heater pipes and connections, aluminum alloy castings for the road-motor department, foundry patterns and aluminum alloy carriage fittings appear to be the main uses. Where aluminum alloy was used in the wrought form for rolling-stock construction, it was only in an experimental stage.

When light alloys are required, the material will be ordered to a British Standard specification.

(g) *Schedule of Non-Ferrous Metals and Alloys.*—For the use of Railway Executive designers, metallurgists, etc., the Committee compiled a "Schedule of Non-Ferrous Metals and Alloys." This was along the lines of the B.S./S.T.A.7 Schedule, and contained details of the chemical composition, mechanical properties, and uses of all the standard alloys.

IRON AND STEEL

A further Policy Committee has been formed to recommend specifications for steels for rolling stock, etc., but, as most of the steels used previously were in accordance with the British Standards, the resulting standardization will not be so spectacular as in the case of some non-ferrous metals. Most grey cast-iron mixtures already include a high percentage of returned scrap, sometimes up to 95%, and further economies are not expected to be great.

RECLAMATION AND RENOVATION OF NON-FERROUS SCRAP

British Railways differ from most other engineering concerns in that their non-ferrous metals are in constant circulation throughout their own system, returning periodically to the workshops either for re-use or for scrapping. It is estimated that 27,000 tons of non-ferrous metal scrap are produced each year by British Railways as the result of machining operations and breaking-up and repairing programs.

It is clear that under such conditions the reclamation and renovation for re-use of as much of this non-ferrous scrap as possible is an economic proposition, as the quantities of expensive virgin metals and alloys which would have otherwise to be purchased are thereby much reduced. It is therefore not surprising that all the former main-line companies carried out in their own workshops a certain amount of reclamation and renovation of those types of scrap which do not have to undergo complicated refining processes or require expensive plant. The L.M.S.R. had developed scrap reclamation to a high degree, and had in being a systematic scheme of scrap collection and segregation, together with a Central Reclamation Depot at Derby, which dealt with non-ferrous scrap from all departments. Established in 1933, this depot is treating annually nearly 4000 tons of non-ferrous scrap; white metal, packing metal, brass and bronze turnings and castings, zinc sheet, lead sheet and pipe, copper wire and cable, white-metal dross, shop sweepings, etc., are all being dealt with. Over 1500 tons of renovated white-metal ingots and 1500 tons of brass and bronze ingots are produced each year. The other Regions all carried out a certain amount of reclamation, but had not developed the process to such an extent as had the London Midland.

Inquiries by the Committee showed that some 80 different varieties of non-ferrous scrap are encountered in Railway Executive workshops and stores. As different terms were being used in the various Regions for what appeared to be the same type of scrap, the Committee recommended that standard terms be used and drew up a schedule listing the type of scrap, how and where it arose, its present description, and the term recommended for future use. These standard terms have now been introduced and are in regular use.

Of the 27,000 tons of non-ferrous scrap produced in Railway Executive shops annually, some 12,000 are sold and the remainder used again by them. Surplus non-ferrous metal scrap and residues are accumulated in the stores at each main works, being segregated according to a standard classification.

The author wishes to thank Mr. R. A. Riddles, the member of the Railway Executive for Mechanical and Electrical Engineering, for permission to present this paper.

The Influence of Specifications on Productivity and the Economic Utilization of Ferrous and Non-Ferrous Metals

BY F. HUDSON OF THE MOND NICKEL CO., LTD.

I shall try to correlate some of the more important aspects associated with the influence of specifications on productivity and the supply position, and to indicate one or two factors of major importance common to the subject as a whole..

Before maximum productivity can be obtained, in conjunction with the most efficient use of available raw materials, consideration must be given to the greater use of simplification, standardization, and specialization methods, so that manufacturing operations can be properly planned. According to the British Productivity Report "Simplification in Industry" these terms can be defined as follows:

Simplification is the process of reducing the number of types and varieties of products made.

Standardization is the process of organizing agreement on (i) a standard for a particular product, range of products, or procedure, and (ii) the application of that standard. A standard is a definition with reference to performance, quality, composition, dimensions, or method of manufacture or testing.

Specialization is the application of particular productive resources exclusively to the manufacture of a narrow range of products.

Simplification and standardization must go hand in hand, since too many specifications may be just as bad as none at all so far as efficient production methods are concerned. An endeavor should therefore be made to plan production so that the minimum number of standard metals and alloys is required. The ideal would be the use of one metal or alloy, a condition which may be difficult to obtain in many factories.

In British Standard 970 (1947), covering wrought steels for use up to 6-in. rolling section for automobile and general engineering purposes, some 137 steels are listed. Are all of these really necessary? British Standard Schedule 1400 (1948), covering copper alloy in-

gots and castings, outlines 23 different alloys for castings. The bulk of industrial requirements for castings can probably be met by 10 copper-base alloys, and these can be readily grouped into 7 main types for production purposes. Agreement regarding tolerances on certain elements, normally considered as impurities, would reduce the number of alloys required still further. The simplification of alloys in this way assists productivity in both melting shop and foundry and aids the subsequent reclamation of scrap. Those responsible for the drafting of future standard specifications should keep this point particularly in mind. It is interesting to note in this connection that the full use of simplification methods has led to a reduction in the number of alloys employed in America. In the plants visited last year by the British Brassfoundry Productivity Team, approximately 75% of the total output of castings was in 85:5:5:5 gun-metal.

Manufacturers should take every opportunity of educating users of metal products to appreciate factors which promote high productivity, namely the use of a limited number of standard alloys and the importance of ordering components of the same design in sufficient quantity, so that mass-production methods may be employed. Users of metal articles should try to apply simplification methods to their products to the utmost possible extent. While it is not always possible to reduce the variety of finished products, many components making up the whole can often be made identical, enabling sub-assemblies of one particular part of the final assembly to be produced in greater quantities.

The next most important factors in obtaining increased productivity in conjunction with the best use of available materials are undoubtedly design and quality control. Specifications in most cases cover only the quality of the metal in a test-bar, and this is of limited value to the engineer in designing a structure composed of rolled, forged, stamped, or cast components. There is particular need for greater attention to be given to the quality of castings, and for research on the relation between test-bar and casting. Designers and engineers today appreciate that neither integral nor separately cast test-bars normally exhibit mechanical properties representative of the casting. When designing castings the engineer therefore increases his section to ensure an adequate factor of safety. This obviously increases the weight of the casting and the amount of metal required. In discussing this matter with a firm of marine engineers of international repute, I was told that the design of bronze and gun-metal castings was based on 50% of the specified test-bar properties. In other words, the castings were twice as thick and twice as heavy as they need have been had the founder been able to produce castings having similar properties to those obtainable from separately cast test-bars. The production of castings of better quality, enabling designers to reduce the section and weight, will obviously lead to substantial economy in the use of metal.

The problem is, however, a long-term one. It is hoped that every brassfounder will give this matter serious attention, as indicated in a paper on "The Mechanical Properties of Some Copper-Base Alloy Castings," presented to the Institute of British Foundrymen by the Technical Committee of the Association of Bronze and Brass Founders in 1949. In that report it is suggested that improvements in the quality of cast products may be sought along several lines. First, by co-operation between designer and manufacturer with a view to evolving designs which will meet the designer's requirements and at the same time be conducive to the production of a sound casting. Secondly, by improvement in manufacturing technique derived from the results of fundamental research, allied with manufacturing experience. Thirdly, by research which

promotes a better understanding of the casting characteristics of the materials, and to provide materials with improved casting characteristics.

Incidentally, the results given in the paper to which I refer support the belief that alloys with long freezing ranges, such as bronze and gun-metal, are liable to contain dispersed porosity which lowers their mechanical properties, while castings in alloys with a short freezing range, such as high-tensile brass and aluminum bronze, can be made completely sound and with properties equal to those of separately cast test-bars, provided the metal is not gassy and that the castings are properly fed. It might be suggested, therefore, that, to make the best use of available supplies of metal, wherever design and service conditions permit centrifugal and chill-casting methods should be used for the production of bronze and gun-metal castings, and greater consideration given to the use of high-strength alloys. This will enable the founder to produce castings with superior mechanical properties, which the designer can utilize for reducing section and weight.

In order to obtain increased productivity from available supplies of metal, it is essential that founders should use metal more economically. This can be brought about by the production of castings of higher quality, enabling engineers and designers to reduce section and weight. Furthermore, this increase in quality must be obtained wherever possible by a reduction in the number of defective castings produced. In order to attain this ideal, brassfounders must give greater attention to the technical control of casting quality. To cover this subject fully would require a separate paper, and at the moment it is possible only to make brief mention of one or two of the more important production factors which merit careful consideration.

These factors include: (i) more general use of degassing methods in conjunction with the development of a rapid control test to assess melt quality; (ii) more effective control of pouring temperatures; (iii) more effective control of the properties of moulding and core sands; (iv) improvements in running and feeding methods; and (v) the adoption of a standard inspection procedure, such as that covered by B.S. 1367 (1947) based on the assessment of actual casting quality, as distinct from the use of separately or integrally cast test-bars. Incidentally, some action on this last point is badly needed, as at the moment manufacturers have to deal with many inspecting bodies, each having its own particular inspection requirements, and this does not assist either productivity or the economic utilization of metals. It would be a great help if we could have one standard inspection system which would be acceptable to all inspecting bodies.

Secondary Heavy Metals

BY E. H. JONES OF CAPPER PASS AND SON, LTD., BRISTOL

Cost of work versus value of metal determines the limits of both primary and secondary recovery, but no minimum concentration of general application can be calculated. Variation in the scale of operations possible and the chemical and physical state of the raw material are very important factors; for instance, the average discard slag of the tin smelter is often higher than the ores before initial concentration. Although it is likely that prices will continue to rise relative to the cost of work, and that the latter will also be lowered by

technical advance, it is possible to make a rough division into recoverable and irrecoverable fractions. Only after a metal has been put to some use can it really be considered secondary, but since the cycling load of residues from smelting and fabricating operations is very large, and as secondary smelters treat considerable quantities of by-products from primary sources, the usual figures given are often misleading.

It is, however, possible to divide consumption into categories for most heavy metals in the U.S.A. and United Kingdom, usually for many years. An estimate of the average life of equipment containing the metals in one form or another can be made from standard depreciation tables and from trade inquiries. Where "old" scrap figures are given in statistics, these can then be related to the period of consumption and a fraction obtained representing what has been recovered (Table XVIII). It is possible to check the results to some extent by comparing those for different metals in the same alloy. The fraction recovered, multiplied by the consumption during the period calculated for the average time in use, will give

TABLE XVIII.—*Consumption and Recovery of Heavy Metals.*

		Element, %	Alloys, %	Coatings, %	Compounds, %	Total Recovered, %
Tin	Used	4	46	48	2	..
	Recovered	2	24	6	0	32
Copper	Used	57	39	..	4	..
	Recovered	38	20	..	0	58
Zinc	Used	7	45	35	13	..
	Recovered	4	25	0	0	29
Lead	Used	40	40	..	20	..
	Recovered	32	28	..	0	60
Antimony	Used	..	60	..	40	..
	Recovered	..	45	..	0	45

TABLE XIX.—*Recoverable Stock of Heavy Metals in Use.*

(Million tons)

	Tin	Copper	Zinc	Lead	Antimony
Author's estimate.....	1.5	40	11	32	0.5
Wegner's estimate.....	..	70	49	70	..

the total recoverable stock under present economic conditions (Table XIX). Wegner has estimated the amount left after deducting loss by corrosion and wear from the total amount mined. His figures are given for copper, lead, and zinc; they appear high, especially that for zinc, but both sets of figures in Table XIX can be taken only as giving some idea of the amounts involved. The differences represent the secondary marginal reserves; an increase in price not only cuts consumption but increases collection from these reserves almost immediately.

The concentration of these stocks is not uniform; the U.S.A. must be increasing its holding as the other countries press manufactured articles on it in return for its highly desirable currency. All figures relate only to the free world.

Consumption, other than stockpiling, including secondaries is:

Tin	180,000 tons
Copper	2,800,000 "
Zinc	2,100,000 "
Lead	2,400,000 "
Antimony	50,000 "

The use of secondary tin in alloys is around 60% of the total, but some of this is derived from the detinning of tinplate. Adjusting for this, it would seem that some 50% of the tin used in alloys is lost. This appears very high, but tin alloys, especially solders, are used in

small portions at a time and are, therefore, well dispersed.

For zinc the figures given for coatings will appear low if referred to published statistics; this is because the tonnage delivered to galvanizers is some 20% higher than that sold on their products, owing to the redistillation of about this fraction from residues, mainly hard spelter, produced during galvanizing. The figure for recovery from zinc alloys is not too sound; it is too early to estimate the fraction for die-casting metal. Brass is the traditional metal for small articles such as screws, fittings, cheap jewelry, etc. This will tend to lower the fraction of alloys recoverable.

COMPOUNDS

It will be seen that salts are considerable fractions and these are almost wholly lost. There are exceptions: for example, zinc used as blue powder as a reducing agent is mostly returned, although it then usually ends as pigment and is finally lost; litharge, if used as paste in storage battery manufacture, is recovered as metal. These uses have not been included in the compound fractions. Normally, a very large proportion is made from metal that has already been used as element or alloy; much low-grade brass is converted into copper sulphate and lithopone. As the price rises, substitution is almost automatic in most cases, but so far no fungicide with all the desirable properties of copper-containing mixtures has been found, and it seems that lead tetraethyl will continue to be used, even with lead at many times today's price, until the Diesel or the internal-combustion turbine replaces the high-compression petrol engine. This use of lead has increased in the last quarter of a century to about 10% of the total consumption and, since the metal is very effectively dispersed, no recovery is possible. If the use continues and increases with greater use of motor transport, it could in the end become the only large-scale use of lead until the adoption of alternative forms of prime-mover became general.

COATINGS

Although plating with nickel and silver is done for the sake of appearance, most coatings are applied to protect a metal with desired structural properties from corrosion. An exception is gilding metal on steel, used in small-arms ammunition, where the backing metal is used mainly to economize in the use of the other. In the case of alloys, the extent of secondary recovery depends very much on the size of the article, since this determines the difficulty of collection after use; but in the case of coatings the general use of steel as the backing metal adds the magnet as a collecting tool, and the proportion recovered is decided by other factors. These are so varied that it will be clearer to deal with the main coated products individually.

Collection of scrap tinplate, including old cans, is well organized, and the return of tin and steel is a significant fraction of the total consumption. As there is no method of removing small amounts of tin from steel, it must be stripped from the plate before melting in the open-hearth furnace. Two processes are used, one depending on the volatilization of tin chloride and the other on the solution of tin in caustic soda. While metallic tin could be recovered by the chloride process, this process is normally used only to the extent that there is a demand for tin chlorides as mordants or reducing agents,

and the metal is ultimately lost. The caustic-soda method yields metallic tin, and is the process used for most detinning.

It will be seen that the operation will be economic only if the value of the coating for unit area exceeds a certain minimum, this in turn being decided by the price of tin and the thickness. Should the value fall too low the scrap would remain untreated and, as the coating may still be too high to be tolerated by the mild-steel maker, both tin and iron would be lost. Assuming that 20% of the tinplate used is recovered when the average coating makes detinning just economical, a cut in the thickness by less than 20% would actually increase the net loss of tin. It is also apparent that a fall in the price of tin would necessitate an increase of coating thickness to avoid the same result; in normal circumstances this would, of course, occur, although there would be a rather long time-lag.

The value of the coating on galvanized iron has never been high enough to warrant dezincing on the lines of detinning operations, but there are also two other significant differences; the coating is to some extent sacrificed during use, and its removal at the steelworks is relatively easy. Zinc used in galvanizing is likely to continue to be lost; if the price rose very high it might pay to treat the material in open-hearth furnaces to make steel and a zincy fume, but the volume of gas to be treated per unit of zinc would be very large. Cadmium plating is mainly electrolytic and, although the price of the metal is very high, the coating is thin; nevertheless, if cadmium-plated articles were in more general use or not so widely dispersed, a leaching method of recovery would be economically possible.

The replacement of cupro-nickel by steel-backed gilding metal (copper 90%, zinc 10%), as with the bullet envelope in small-arms ammunition, brought recovery problems. Since only bullets collected at practice butts are recovered, the saving of copper and nickel must have been considerable, but the scrap strip from which the blanks had been punched, and the wasters, could no longer be re-melted and re-rolled. Although during the war at least two processes were recommended specifically to separate and recover the copper and iron, most of the scrap was used as flux in lead smelting with the formation of a leady copper matte, the copper ending as copper sulphate and the iron as silicate in the slag. There is no doubt that, with an assured supply warranting the investment in plant, either of the two processes mentioned above would have been much more efficient, but war is still considered a temporary state of affairs.

ALLOYS AND ELEMENTS

Except in the case of elemental zinc, corrosion and wear are probably not the cause of serious loss. Indeed, not all the products of corrosion are lost; they may adhere to the surface, or be collected, as in the case of lead sulphate from chemical plants, the metal content being recovered by the secondary smelter.

The proportion ultimately recovered of the metal used as the element, or as an alloy of which it is a major constituent, depends mainly on the size of the article manufactured. A very easy method of conserving, say, copper would be for the United Nations to insist that no end-use product of copper, brass or bronze should weigh less than a pound. The effect would be chaotic, but there is no doubt that the method would appeal to modern governments much more than the traditional one of allowing the price

to increase. Copper and zinc in screws, nuts and bolts, locks, keys, furniture fittings, and such small items must be written off as almost wholly lost.

The collected scrap can be divided into two main categories: (i) clean pieces that can be re-melted and, after adjustment of composition, cast into ingots for return to the fabricator; and (ii) material that by design or accident has become attached to other metals or inorganic materials so firmly that physical separation is difficult or impossible. The latter are smelted, and during the operation some of the metal is lost. Sometimes in the case of alloys the whole of one constituent may be wasted; for example, the presence of steel inserts, screws, etc., in brass articles has in the past caused considerable dispersion of zinc into the atmosphere. Even today, when most copper smelters' fume is collected, a process for the removal of iron from brasses and bronzes would save much metal and an even greater value in metallurgical work.

Smelting removes non-metallics, aluminum, magnesium, and some zinc in the slag, and volatilizes zinc and cadmium which are caught as an impure fume. The metallic product will normally be an impure mixture and, if this can be refined and adjusted to a marketable alloy, the loss and work will naturally be less than if the elements have to be produced to ensure sale. The alloy must normally meet some specification or other and, if the maximum metal is to be regained from scrap, it is essential that such specifications should be realistic. A very low limit put on, say, iron or nickel in brass, bronze, or solder, or on bismuth in lead alloys, because it is safer in the absence of information about the amount that could be tolerated without deleterious effects, wastes both work and metal.

The introduction of new alloys containing elements that were not previously used in the same range can increase the problems of the secondary smelter very considerably. Even if the greatest care is taken in segregation during collection, some mixing of scrap and residues is inevitable, and the abnormal elements must be removed by metallurgical means before an alloy of use as a basis for a salable range can be obtained. Examples are the use of nickel in bearing metals, tellurium in lead, and arsenic and silver in solders.

The effect of relative prices not only of the metals but of fuel, power, labor, etc., on the adjustment of the smelters' flowsheet, is considerable. When the price of zinc is low, the lead smelter will use it to remove impurities and ultimately lose it in slag; aluminum will be used to remove antimony from solder, instead of the more laborious process of liquation or fluoborate electrolysis. The optimum economic metal content of discarded residues, whether the slag of copper, tin, and lead smelting, or the retort residue in the case of zinc, is a varying equilibrium between value of recovery and cost of work done, and is the subject of constant calculation. It has been truly said that a smelter should never finally throw away anything, since the concentrations in Nature are being steadily exhausted and the relative value of the metal increasing. Dumps of slag and residues then become valuable reserves.

The secondary smelter very rarely treats only materials that have been used by the community; much of its raw material consists of complex concentrations of by-products from primary plants, and quite often complex minerals as well. The plant must be as flexible as possible and, therefore, the tonnages produced relative to the size of the enterprise seem very small compared to those of the primary smelter. The great variation in feed makes the production of a range of elements and alloys necessary, although usually these are kept to a minimum by selling by-products to other

refiners. A convenient and competitive outlet for materials not suitable for treatment at the producer's plant is of great value. A comprehensive extractive metallurgical industry, comprising many plants each specializing in some field, was an important factor in the success of German pre-war secondary smelting.

There is scope for very considerable technical improvement; the treatment of complex arsenides is alone worth study, but it is doubtful whether the difference between present technique and perfection would materially alter the rate at which man is scattering the metals under review beyond practical recovery. This need not worry us unduly; for most uses there are economically possible substitutes for each of these metals: reserves, primary and secondary, are sufficient to supply the remaining uses for a time stretching well beyond that within our power to foresee.

Secondary Aluminum and Magnesium

BY COLONEL W. C. DEVEREUX, C.B.E., MANAGING DIRECTOR, ALMIN, LTD.; CHAIRMAN, INTERNATIONAL ALLOYS, LTD.

The production of secondary light metals and alloys from scrap and waste materials, as an integrated and scientifically controlled industrial process, was somewhat delayed in this country. In the U.S.A. and Germany, the basis of such an industry had been laid during the 1914-18 war, and by 1926 secondary aluminum in the U.S.A. had reached an estimated annual production of 44,000 tons.

The United Kingdom secondary aluminum industry as we know it today did not really get under way until the early thirties. In those days the consumption of aluminum in the United Kingdom was relatively small—about 20,000 tons of primary metal a year, out of a world total of 180,000 tons. Moreover, consumption in earlier years had been even smaller, so that the arisings of old scrap had not then been large enough to sustain a separate secondary industry. Once started, however, the secondary smelting industry advanced rapidly. At first the smelters had to content themselves with supplying second-grade foundry alloys and deoxidizing material and with exports; but by 1935 they had reached a production of about 10,000-12,000 tons a year.

By a vigilant policy of laboratory control of quality, they patiently set about overcoming the prejudices against their products, and by the outbreak of the recent war it was generally accepted that secondary alloys were metallurgically equivalent to those made from primary material. The advent of the war brought about a complete change in the status of the industry, which now assumed a role of great importance and, although the discriminatory clauses were not removed from official specifications until after the war, they were tacitly ignored. Table XX shows production and consumption since 1940, when 32,000 tons of secondary metal were used. Since the war its use has continued to increase, reaching a record yearly rate of about 130,000 tons in the first half of 1951—over a third of the total raw material

used by the aluminum fabricating industry. In terms of money this represents about £14 million.

The recovery of magnesium alloys from scrap had meanwhile also made progress. No statistical data are available, but it is estimated that of the total consumption, which at present is at the rate of 5500 tons per year, something between a quarter and a third is met by recovery from scrap.

TABLE XX.—*Production and Consumption of Secondary Aluminum in United Kingdom.*

Year	Secondary Production, tons			Aluminium Consumption, tons			Percentage of Secondary Used
	Smelters	MPRD's*	Total	Virgin	Secondary	Total	
1940	37,000†	900†	37,900	102,300	32,200	134,500	24
1941	50,300	2,900	53,200	116,800	49,000	165,800	30
1942	75,300	3,500	78,800	195,300	63,800	259,100	25
1943	87,700	5,800	93,500	208,200	85,500	293,700	29
1944	95,900	8,600	104,500	150,100	71,700	221,800	32
1945	61,900	19,100	81,000	94,900	51,200	146,100	35
1946	48,600	26,300	74,900	114,600	82,600	197,200	42
1947	63,700	33,900	97,600	159,200	107,500	266,700	40
1948	63,300	16,800	80,100	173,700	69,400	243,100	29
1949	68,600	6,600	75,200	178,600	68,400	247,000	28
1950	86,400	..	86,400	181,400	93,800	275,200	34
1951‡	89,300	..	89,300	185,300	131,800	317,100	42

* Ministry of Supply Recovery Plants.
† Estimated.
‡ Estimated as twice the figure for January-June 1951.

SORTING OF SCRAP

Hand sorting of large pieces of scrap is still done mainly by experienced but not technically trained workers, with the support of a few very simple tests. The so-called spot tests for identification of alloys have not found use as routine tests in sorting single pieces, as they are still too slow for this purpose. Recently, instruments have been developed which seem to offer possibilities of comparing in a few seconds the electrical and magnetic properties of alloys. Perhaps these instruments will in future become useful in routine sorting in special cases. A very striking improvement in sorting efficiency has been effected by one company by putting a skilled chemist on the job to guide and support the sorters with various tests and better general technical knowledge.

Sorting of small pieces can be economical only if it is mechanized, and so far is limited to magnetic separation of ferrous metals. Obviously there is great scope for further inventions in this field. Large-scale experiments with mechanized sorting on the sink-and-float principle were technically successful, but the present market situation does not permit exploitation of the process.

For composite structures containing light metal parts and other metals, sorting before the melting operation is often impracticable, and separation is effected by the use of liquation furnaces, a method which has been used on a large scale for aircraft scrap. As the use of the liquation furnace seems now firmly established, it will be interesting to compare views on the most suitable designs.

MELTING AND METALLURGICAL TREATMENT

On the question of furnace equipment there is at present a division of opinion about the merits and limitations of the rotary furnace, which is built with capacities between 1 and 8 tons, and reverberatory furnace, which generally covers the range between 10 and 25 tons capacity. The Americans, with their generally larger output and their strong interest in mechanizing charging and pouring operations, have set themselves to adapting the reverberatory furnace to intensive flux treatment as necessary for the melting of scrap, and have produced very useful furnace designs. On the other hand, in Europe the rotary furnace, with its short melting cycle and great flexibility, is firmly established. The two methods have been compared recently by a mission of European specialists sent to America by O.E.E.C., and it is hoped that a useful clarification of opinions on this point will arise from their reports.

A recent development in the melting equipment is the use of low-frequency induction furnace for melting many kinds of scrap, including turnings, as long as they are fairly clean. This equipment has been operated mainly in Germany and America, and apparently has given good results in production.

Refining of aluminum and magnesium in the classic sense, i.e., for production of pure aluminum or magnesium, is at the present time technically possible but not economically justified, since, as a rule, all the metallic material available can be incorporated into useful alloys. The scope of the present refining technique is, therefore, confined to the elimination of metallic inclusions, oxide, etc., and of gases, the treatment consisting essentially of melting under a flux the action of which is to lift the oxide from the liquid melt by surface-tension forces and to keep it enveloped in slag. It has been shown that solution of oxide in the slag does not play any appreciable part in the slag treatment either of aluminum or of magnesium.

The only industrial refining operation in the more classic sense is the elimination of magnesium from aluminum alloys. The removal of other alloying elements is not at the moment economical.

High-class technical, and especially analytical, laboratory control is essential in all stages of scrap recovery, sorting, blending, furnace control, and release. Requirements in precision of analysis in secondary-metal production are often higher than in producing alloy from virgin material. As a consequence, the art of analysis, especially of aluminum, has received valuable contributions from the work of the laboratories of the remelting industry.

The production of secondary magnesium alloys proceeds technically on similar lines to the production of secondary aluminum alloys, with the well-known differences in fluxing and melting treatment. This applies to the more compact forms of scrap. Turnings and sawings can be re-melted with due precautions, but, apart from the flux consumption, the economy of this recovery is reduced by the precautions to be taken in storing and transporting this material, which is more sensitive to the influence of air and moisture than the corresponding aluminum scrap. The recovery of magnesium from skimmings and grindings is not practicable at the moment.

SCRAP ARISINGS

By no means all the scrap metal recovered is handled by the secondary smelters. There is always in the works of the fabricators a large amount of scrap runners and risers, discards, sheet clippings, etc., which nowadays never comes on to the market at all, but which goes straight back into the furnaces with the primary material. Material of this kind is termed *process scrap*.

New scrap is scrap arising at the second stage of manufacture, i.e., in the works of the users of semi-manufactures. The quality of this scrap ranges from fairly massive and therefore readily identifiable forms, such as off-cuts from sheet and extruded sections, and scrapped castings, to machine-shop swarf and turnings. Not all this is available to the secondary industry, owing to special arrangements sometimes made by fabricators with their larger customers for the return of the more solid types of scrap. Since the turnover of this scrap is fairly rapid, the quantity available bears a fairly constant relationship to the scale of activity of the fabricating industry. This is true also of dross, skimmings, grindings, etc., which, although strictly coming under the heading of new scrap, require such specialized treatment that they constitute a separate class of their own, and are almost all handled by the remelting industry.

The third main group of scrap is *old scrap*, i.e., arisings from obsolete, damaged, or worn-out machinery, structures, utensils, etc. It also includes the special class of aircraft scrap.

The question of what proportions of our secondary metals derive from new and old scrap respectively is interesting and important to the study of future trends. New scrap arisings are directly related to the scale of use of aluminum at any given time and bear a fairly constant relationship to it, although this proportion is not fixed absolutely. If anything, it tends to fall as efficiency of utilization increases, a point well illustrated by the reduction in the proportion of turnings arising as the use of die-castings increases at the expense of sand castings. Unfortunately, we have no accurate statistics on this point, though we may learn something from the U.S. figures shown in Tables XXI and XXII.

TABLE XXI.—*Production of Secondary Aluminum in U.S.A.*

(Short tons)

Year	From New Scrap	From Old Scrap
1935–1939 (average)	16,000	35,000
1939	16,200	37,800
1940	34,600	45,800
1941	63,700	43,100
1942	154,800	41,600
1943	280,900	33,100
1944	302,700	22,900
1945	271,100	27,300
1946	187,500	90,500
1947	181,000	163,800
1948	191,100	95,000
1949	136,200	44,600

It will be seen that in the pre-war years about 70% of secondary metal in the U.S.A. derived from old scrap, but that since then the proportion of new

TABLE XXII.—*Consumption of Scrap by Remelters, Smelters, and Refiners in U.S.A., in 1949.*

(Short tons)

Source	New Scrap	Old Scrap
Pure clippings, wire foil, etc.	9,618	273
Castings, forgings	14,310	12,027
Alloy sheet	21,027	2,511
Scrap sheet, sheet utensils		5,900
Borings and turnings	27,532	
Wrecked aircraft		10,188
Miscellaneous and dross	12,798	12,288
Total	85,285	43,187

scrap has exceeded old scrap by a considerable margin. This is to be expected, since in 1935-39 the fabrication of aluminum was increasing only slowly, while expansion since then has been very great indeed. The really interesting thing about these figures is the trend of the old scrap figures. There is the effect of the scrap drive in 1940-42, and the shortage in 1943-45 as a result thereof. Note also the great increase in 1945-48, when aircraft scrap was used in large quantities, and the apparent return to normality in 1949.

SUPPLY AND DEMAND

Generally speaking, the relation of supply to demand for scrap and secondary metal follows the same lines as for other basic commodities, being perhaps among the items most sensitive to economic conditions. While production of fabricated products rises, scrap arisings tend to fall behind demand because of their relation to the smaller scale of earlier production. Conversely, it would appear that a surplus of scrap should arise when production decreases, because the process scrap from previous larger output flows back to be combined with the present smaller production as shown in Table XXIII. It must be admitted that this is not borne out by the events of the past six years, but then special measures were adopted to utilize the accumulated aircraft scrap, for instance, by inclusion in the aluminum houses of which vast quantities were sold, much to our regret at the moment.

TABLE XXIII.—*Aluminum Scrap Arisings and Consumption in the United Kingdom.*

Year	Arisings, tons	Consumption, tons	Total Consumption of Virgin and Secondary,* tons	Arisings as Proportion of Virgin and Secondary Consumption, %
1941	66,700	59,400	165,800	40
1942	97,300	91,700	259,100	38
1943	121,800	110,900	293,700	41
1944	125,400	123,500	221,800	57
1945	71,000	83,200	146,100	49
1946	73,700	70,200	197,200	37
1947	118,300	92,300	266,700	44
1948	131,100	102,100	243,100	54
1949	99,900	104,500	247,000	40
1950	92,800	118,500	275,200	34
1951†	93,200	110,300	317,100	29

* This total is somewhat in excess of actual consumption, since secondary output contains some virgin metal which is therefore included twice. However, although in the present circumstances this is higher than usual, the proportion of virgin metal is normally well below 10%.

† Estimated as twice the figure for January-June 1951.

In an interesting article written in 1944, Korn calculated that there may come a time in the distant future when, under stable conditions of aluminum consumption, 75% of the total demand for aluminum could be supplied from secondary metal. We are not concerned here to look so very far ahead, but it is a suggestion of how the importance of secondary metal will tend to increase. At present in most industrial countries the proportion of secondary metal, excluding process scrap, incorporated into the total aluminum consumption tends to be around 30%, as shown in Table XXIV.

Over-all United Kingdom production of secondary ingot since 1940 has been in excess of consumption, so that substantial stocks were built up, amounting to 118,000 tons at the end of 1949. In the last year, however, consumption has increased sharply and currently stocks are being consumed at the rate of about

TABLE XXIV.—*Proportion of Secondary Metal in Aluminum Consumption of Various Countries.*

(In 1000 metric tons)

	Total Primary and Secondary Consumption			Secondary Consumption			Per cent. Secondary		
	1948	1949	1950	1948	1949	1950	1948	1949	1950
U.S.A.	883	742	1082	261	164	265	30	22	24
Total O.E.E.C. countries	538	490	566*	163	140	170*	30	29	31
United Kingdom	245	250	278	69	69	94	28	28	34
France	104	68	74*	25	20	20*	24	29	27
Germany	75	59	75*	42	23	30*	56	39	40

Sources: For European countries, O.E.E.C. reports. For United States, Bureau of Mines. For United Kingdom, Ministry of Supply Light Metal Statistics.
* Estimated.

40,000 tons a year, at which rate stocks will be exhausted by the end of 1952, if not before.

A similar situation exists in the case of scrap arisings. Excluding arisings of aircraft scrap consumed at government scrap-recovery plant, the excess of arisings over consumption from 1941 to July 1951, was about 30,000 tons. At the current rate of excess of consumption over arisings of about 20,000 tons a year, this source will soon be exhausted. If the rate of scrap arisings were to remain as at present, i.e., about 95,000 tons, then the production of secondary aluminum would be limited to about 82,000 tons, i.e., about 8,000 tons below the rate in the first half of 1951 and nearly 50,000 tons below the current rate of consumption. These figures must not be taken too literally, as experience has shown that more scrap can usually be found when the time comes to "scrape the barrel"; but it is certain that even if the smelters can somehow maintain the present rate of output, the amount available for consumption must fall considerably.

What makes attempts at accurate forecasts very unreliable is the dynamic nature of such a young industry as ours, particularly in regard to the constantly changing pattern of the uses of aluminum. Before the war a very high proportion of the aluminum used went into machines of moderate life-span, e. g., motor vehicles, aircraft, etc., whereas some important post-war applications have been developed in which aluminum was chosen partly because of the very long life-span it offers. In particular, I would mention building applications, which have consumed about 250,000 tons since 1945. It will be many years before material so used will come on the market as scrap. Such uses are, however, balanced to some extent from the secondary point of view by a rapid rise of some very short-life uses, particularly in the packaging industry, but to exploit this potential source of scrap presents very considerable difficulties.

Since 1948 the use of aluminum for packaging has more than doubled. It is currently running at about 30,000 tons a year. Assuming 70% of this actually goes into food cans, bottle lids and caps, foil wrappings, etc., there is some 20,000 tons of potential scrap which should arise within a year or two of manufacture. This scrap is the most difficult to collect, relying mainly on collection by local authorities, charitable and voluntary organizations, hospitals, etc. However, this is potentially worth something like £1½ million a year, and seems likely to increase, so that it might be worth while considering expenditure of a few thousand pounds a year in propaganda to make the public aware of the importance of this and other forms of household scrap as dollar savers and to encourage their segregation and collection.

INFLUENCE OF EXPORT DRIVE

The high rate of export of manufactured goods has a very profound influence on the arisings of scrap, since many of the commodities for which there are the largest export markets use large quantities of aluminum, e.g., motor vehicles, vacuum cleaners, washing machines, etc. Road vehicles, for example, are among the largest users of aluminum alloys and before the war scrapped motor cars and components provided substantial quantities of material for secondary smelters.

Road vehicles consume nearly 50,000 tons of aluminum semi-manufactures a year. If we assume that 80% goes into the finished vehicle, then almost 40,000 tons is sent out of the country every year, leaving only 10,000 tons as potential arisings when the vehicles are scrapped, although, of course, with the present shortage of vehicles at home, we shall have to wait a long time for this. The actual arisings now are but a fraction of this figure, being about a quarter of the arisings from this source before the war, in spite of a 100% increase in vehicle production.

From these considerations it appears that we must anticipate a shortage of scrap during the next few years. The abnormally high rate of consumption of old scrap made possible by the channeling of all aluminum into short-lived structures during the war, the necessity to limit dollar imports, the high rate of indirect export of aluminum, the development of long-life uses of the past few years, the difficulty of collecting short-life scrap, and the current rearmament program all tend to reduce the scrap arisings or increase demand.

In the long run, the more virgin aluminum that is used the greater the potential scrap arisings, but in the immediate future it is difficult to see how a falling off in supplies can be avoided unless drastic steps are taken to improve the efficiency of scrap collections and utilization. This is not a matter which concerns the scrap merchants and secondary smelters alone. As previous speakers have emphasized, the utilization of scrap is an important factor in the national economy. Improved utilization requires the wholehearted cooperation of the fabricating industry and all users of light metals in scrap segregation and collection and the elimination of wasteful methods of remelting.

DESIGN FEATURES OF A 250-KW GAS-TURBINE ENGINE

ACKNOWLEDGMENT

This article describes a 250-KW gas-turbine engine used to drive a shipboard emergency generator. The authors are R. R. Peterson of the Bureau of Ships, Navy Department, and P. G. Carlson of Solar Aircraft Company. The paper was presented at the Annual Meeting of The American Society of Mechanical Engineers, November 25-30, 1951, and was published in the March 1952 issue of "Mechanical Engineering."

CHARACTERISTICS

In its present state of development, the simple-cycle, non-regenerative, gas-turbine engine is ideally suited to applications where weight and space must be kept to a minimum and where the utilization factor is low, making fuel consumption a secondary factor. An application having these general requirements is found in the gas-turbine engine as a prime mover for a shipboard emergency generator set. In addition to the generally recognized advantages of reduced size and weight, a gas-turbine-engine generator set for naval use offers certain other potential advantages over conventional reciprocating engine sets which are as follows:

(*a*) A smaller number of parts simplifies the maintenance and spare-parts problem.

(*b*) The set can operate without cooling water.

(*c*) The high-frequency noise can be attenuated significantly by a simple means.

By 1947 the practicability of the gas-turbine had been established, largely in the field of aircraft propulsion, and the conclusion was reached by the Department of the Navy, Bureau of Ships, that the advantages offered by the gas-turbine-engine emergency generator as outlined warranted development of such a unit. Accordingly, specifications for competitive bids were issued with the requirements as given in Table 1.

In order to utilize new developments and configurations to the greatest extent, no restrictions on design features or methods of construction were imposed. The present gas-turbine engine for an emergency generator set was planned for installation in a ship being constructed by the Bath Iron Works Corporation with Gibbs and Cox, Inc., New York, N. Y., acting as design agent. In June, 1947, a Gibbs and Cox purchase order for this engine (designated the T-400) was awarded to Solar Aircraft Company, San Diego, Calif.

TABLE 1.—REQUIREMENTS FOR GAS-TURBINE-ENGINE EMERGENCY GENERATOR SET.

Electrical output, kw	250
Air-inlet temperature, deg. F.:	
Normal	80
Maximum	100
Exhaust back pressure, in H_2O max.	37
Fuel	Standard Diesel oil
Starting medium	Compressed air or cartridge
Starting time, standstill to full load, sec. max.	10
Control	Automatic starting and operation, starting signal provided by loss of voltage from main ship's service generator. Means also to be provided for manual operation.
Speed governing:	
Steady state, percent	± 0.5
Transient, percent	± 4.0
Recovery time, sec. max.	0.025
Life	5000 hr of which 500 hr shall be at full load
Sound level	100 decibels (0.0002 dynes per sq cm).
Maximum inclination:	
Permanent	5 deg from horizontal, fore and aft, and 15 deg from horizontal, to either side.
Rolling	45 deg from vertical, to either side.
Pitching	10 deg from horizontal plane.

The basic gas-turbine engine was completed and initially run in September, 1949. Following a development program during which certain design changes were incorporated and accessories and controls were completed and tested, the power plant was delivered in June, 1950, for further testing at the U.S. Naval Engineering Experiment Station, Annapolis, Md. Operational testing at the Experiment Station was begun in August, 1950, and concluded in February, 1951, at which time the gas-turbine engine was delivered to the shipyard for installation.

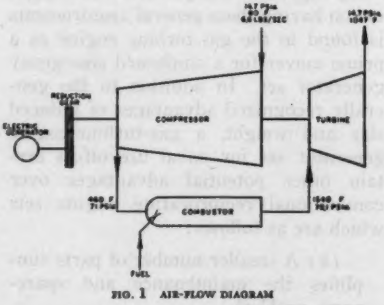


FIG. 1 AIR-FLOW DIAGRAM

The air-flow diagram, Fig. 1, shows the sequence of components in the gas-turbine cycle, together with the pressures and temperatures existing when the generator set is providing rated continuous output with ambient-air conditions of 80 F and 14.7 psi pressure. The air-flow rate is approximately 6.6 lb. per sec. Fig. 2 shows fuel consump-

tion and turbine-inlet temperature at various loads and ambient temperatures. The specific fuel consumption is seen to be approximately 1.07 lb per hphr based on shaft power to the generator.

The design pressure ratio of 4.8 and maximum gas temperature of 1540 F were selected to provide a reasonable compromise between cycle efficiency, life of turbine components, and power-plant size. Considerations of adiabatic efficiency at the pressure ratio chosen, compactness of arrangement, and minimum development time dictated the use of a 10-stage, approximately 50 percent reaction, axial-type compressor. The comparatively small size of the compressor (9.5 in. average rotor diameter) with associated scale or size effects arising from relatively larger blade-tip and seal clearances and reduced Reynolds numbers, results in a compressor efficiency somewhat less than that obtained on larger axial-flow compressors for which data are available. The peak adiabatic efficiency of 85 percent and design point efficiency of 80 percent are, for the pressure ratio selected, still substantially better than could be obtained with other types of compressors on which scale effects may be less. At rated speed of 20,138 rpm, compressor blade-tip velocity is slightly over 900 fps.

Three or four turbine stages would be required to obtain optimum turbine efficiency with the pressure ratio selected. However, in the interests of simplifying the unit and keeping size and weight to a minimum, two stages were chosen, resulting in somewhat poorer turbine performance than otherwise could be obtained. The turbine blading is designed on the vortex principle, utilizing 20 percent and 11 percent reaction at the hubs and 49 and 48 percent at the tips of the first and second stages respectively. Tip speed of the turbine under rated conditions is 1054 fps.

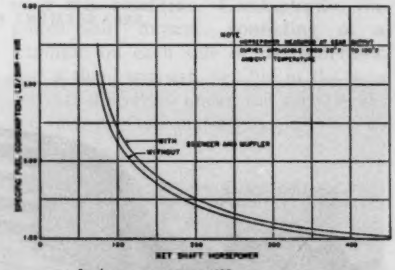


FIG. 2 PERFORMANCE OF T-400 GAS-TURBINE ENGINE

Pressure drops through the inlet silencer and the exhaust muffler of approximately 4 and 8 in. of water, respectively, impose a penalty of some 20 hp on the engine output. Sound level at a point 6 ft from the power plant and directly in line with the compressor inlet was found to average 97 decibels, a reduction of 10 to 12 decibels from the value obtained without the silencer. Use of the exhaust muffler did not influence appreciably the sound level immediately adjacent to the power plant; however, at exhaust-stack discharge the sound level was reduced from approximately 100 to 95 decibels.

Compressor surge was found to limit the amount of fuel that could be injected during the starting cycle. For this reason the starting-time objective of 10 sec maximum was not attained. The starting time obtainable at present, from starting signal to full speed, is approximately 17 sec. This probably could be reduced by bleeding off compressor air, or by installing a by-pass around the turbine first stage. The complications that would arise from such changes, however, do not appear to be warranted. It should be noted that starting time is a function of the mass and configuration of the driven equipment as well as the energy available for turbine starting.

ARRANGEMENT OF COMPONENTS

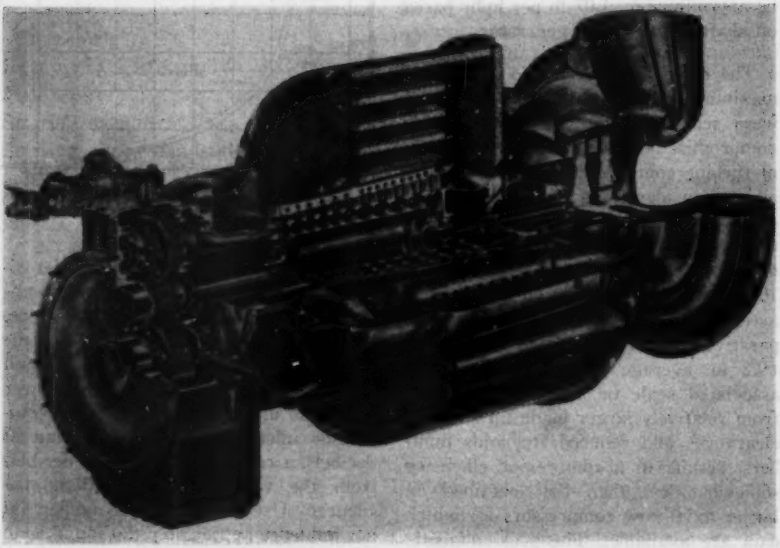


FIG. 3 CUTAWAY VIEW OF T-400 GAS-TURBINE ENGINE INCLUDING INLET SILENCER

The mechanical arrangement of the various components is illustrated in Fig. 3, a cutaway drawing of the unit, and Fig. 4 the assembled unit. It will be noted that the shafts of the principal components (reduction gear, compressor, and turbine) are connected by splined couplings. Inlet air is inducted through a combination silencer-and-oil-cooler assembly located near the center of the engine, then flows forward through the compressor, is diffused, reversed 180 deg in direction, and passes into a single combustion chamber paralleling the compressor. The hot gas produced in the combustor enters a turbine scroll which conducts and distributes it to the first-stage turbine nozzle. At the exit of the turbine the gas is diffused, after which it enters an exhaust collector which discharges vertically upward. An exhaust muffler is located in the exhaust stack.

It was recognized that a significant pressure loss would be associated with reversing the compressor discharge and locating the combustor parallel with the compressor. However, this arrrangement shortens the power plant more than 15 percent, and, therefore, is considered worth while since length is the critical dimension in this application.

Accessories mounted on and driven from the gearbox include the governor, lubricating-oil pump, fuel pump, speed-sequencing-switch assembly, and tachometer generator.

Considerations of the stresses imposed by shock loads indicated that the loads could be absorbed without the use of shock mounts and without significant weight penalty, providing the generator chosen for the installation could also meet this condition. Consequently, the power-plant mounts, consisting of a trunnion on each side of the generator and a third support, flexible in the fore and aft direction, under the turbine, do not incorporate isolating media.

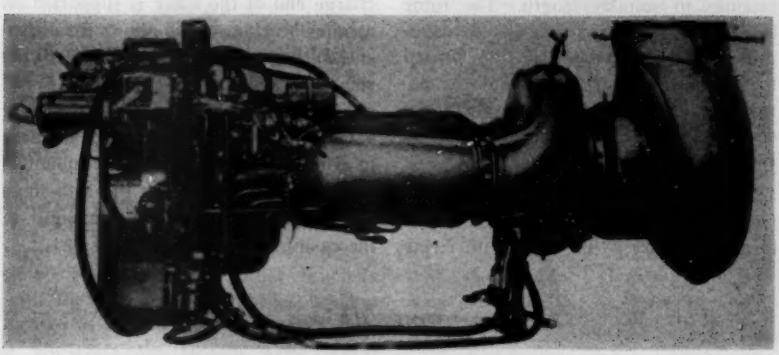


FIG. 4 T-400 GAS-TURBINE ENGINE

The weights of the principal components are as follows:

		Pounds
Gas-turbine engine, including ignition system, filters, and piping		350
Control box and instrument panel		56
Reduction and accessory drive gearbox		130
Driven accessories (fuel and lube pumps, governor, etc.)		25
		561
Inlet silencer and exhaust muffler	260	
Total with silencer and muffler		821

Outline dimensions are length 59¾ in., width 33 in., height 34 in.

COMPRESSOR

Fig. 5 shows the compressor rotor. A constant 8-in. rotor-hub diameter (rather than maintaining constant blade-tip diameter) is employed to simplify manufacture, particularly in machining the blade attachment slots, although an additional stage is thereby made necessary to obtain the design pressure ratio. This arrangement results in somewhat greater blade heights in the high-pressure stages,

FIG. 5 COMPRESSOR ROTOR

which is advantageous performancewise in a small compressor.

The rotor blades, made of type 403 stainless steel, and stator vanes, made of type 347 stainless steel, are produced by cutting four designs of investment castings to suitable length. The rotor blades are attached by means of dovetail joints. The stator vanes are brazed to inner and outer rings, and the assemblies are bolted to the cast-aluminum compressor casing which is made in two 180-deg segments.

The compressor disks are machined from type 403 stainless steel and secured to a drum of 5 5/16 in. diam with rivets which extend radially through the overlapping flanges of the disks and through the drum.

The compressor end thrust of almost 600 lb is absorbed by a single ball bearing located at the inlet end. The discharge end of the rotor is supported on a roller bearing, which allows for differential thermal expansion between the rotor and stator. The compressor scroll casing carries outlet air heated to about 450 F by compression. To cool the area in which the roller bearing is mounted, a small fan integral with the 10th-stage disk circulates air through passages in the casing.

COMBUSTOR

The combustor is of the conventional, through-flow, can-type design with perforated liner such as is used extensively in aircraft turbojets. Specific-heat release at full load is approximately 5×10^6 Btu per hr per cu ft per atm.

Use of a single combustion chamber eliminates the possibility of starting with an unignited burner. The single chamber is easily removed and inspected without disassembly of the power plant. With injection of fuel through only one nozzle, the nozzle orifice can be kept large, and hence is less apt to clog from impurities in the fuel.

In order to achieve good fuel atomization throughout the speed range and thus assure consistent firing, a variable-area fuel nozzle is employed. A conventional gas-turbine spark plug, energized by a storage battery and vibrator coil during the starting cycle, provides ignition.

TURBINE

The maximum gas temperature chosen, in conjunction with the conservative design stresses, does not require extensive cooling of the turbine parts. The alloys selected for the critical parts, S-816 and N-155 for the rotor blades and disks, respectively, and Hastelloy C for the nozzle vanes, provide creep and rupture life expectancies of such duration that only moderate cooling of the turbine disk face is required. A small fan integral with the first-stage disk circulates cooling air over the hub and upstream face of the disk. This air then passes around the turbine-inlet scroll, through its shrouding, and around the turbine casing, discharging into the turbine exhaust. A second fan, mounted on the downstream side of the second-stage disk, circulates cooling air in that region.

Conventional fir-tree joints are used to attach the turbine blades to the disks. An unusual feature of the attachment, however, is the loose fit employed. When inserted into the disk slots the tips of the blades can be moved 0.020 to 0.030 in. in the tangential direction. Before the radial retaining pins are installed, the blades will drop out if the rotor is held with the axis vertical. The loose fit allows for differential expansion between the serrations on the blade roots and disk slots and permits the blades to seek positions minimizing the bending stresses arising from centrifugal loads. The chrome-molybdenum steel shaft is arc-welded to the first-stage disk while the second-stage wheel is attached with three ½-in. bolts. The turbine rotor is supported on ball and roller bearings with the extra-clearance roller bearing located at the turbine or hot end. Fig. 6 illustrates the turbine rotor (first stage).

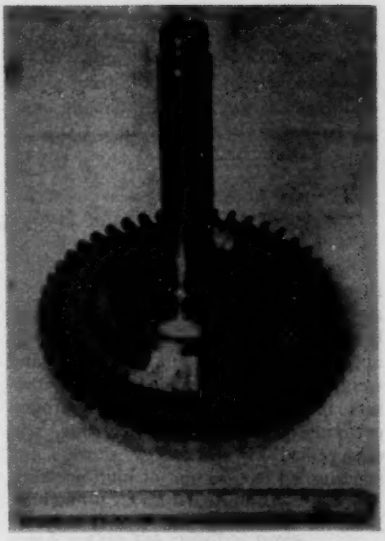


FIG. 6 TURBINE ROTOR

REDUCTION GEAR

A single pinion-and-gear combination reduces the speed from 20,138 rpm to that required by the generator. A set of accessory gears driven from the input pinion provides drives at 3358 rpm for each of the four accessory mounting pads, located for accessibility at the top of the gearbox.

LUBRICATION SYSTEM

High-speed antifriction bearings in gas-turbine service must be cooled as well as lubricated. One common method of cooling consists of directing upon the bearing a jet of compressor bleed air, to which a small amount of lubricating oil is added. Another method involves increasing the lubricating-oil flow to a point where it acts as a coolant as well as a lubricant. The latter method has been used widely with success in the field of aircraft turbosuperchargers and gas turbines, and on that basis was selected for this application. Results to date have been entirely satisfactory.

A 3.5-gpm vane-type pump draws oil from the 4-gal reservoir located in the reduction-gear casing, and pumps it through a micronic filter and an oil cooler consisting of a bank of finned tubes located in the air-inlet-silencer assembly. From the cooler, oil is distributed to the lubrication points in the gearbox and to the compressor and turbine shaft bearings, each of which receives approximately 1 qt per min of oil. The oil used is Navy Symbol 2075.

Transferring its heat to the compressor-inlet air may seem a costly way to cool the lubricating oil in view of the

gas turbine's pronounced sensitivity in performance to inlet-air temperature; however, the relative magnitudes of the compressor air flow and the lubricating-oil heat rejection are such that the inlet-air temperature rise due to oil cooling is not prohibitive.

A scavenging element in the same housing with the oil-pump pressure element returns oil from the turbine rotor and aft compressor bearings to the reservoir, while the oil from the forward compressor bearing and gear trains returns to the reservoir by gravity.

STARTING SYSTEM

The large amount of starter power required to provide the specified rapid acceleration ruled out a conventional battery-driven electrical starter for this application because of the size and weight involved. Consideration of factors involved in the use of hydrogen peroxide, steam, solid propellants, and compressed air as energy sources for starting led to selection of high-pressure (3000 psi) compressed air as a medium. Ease of handling and availability of compressed air on board ship also influenced the selection. The air is stored in a tank of sufficient size to provide five normal starts. A pressure regulator reduces air pressure to 400 psi, after which it is directed at the second-stage turbine blading through a number of small jets. Although the power plant is self-sustaining at a lower speed, starting assist is maintained up to 55 percent of rated rpm to provide rapid acceleration in the range where the margin of turbine power over compressor power is small.

FUEL AND CONTROL SYSTEM

A large portion of the developmental effort was involved in providing a control system which would fulfill the following requirements in the absence of operating personnel:

1 Initiate the starting cycle when normal ship's voltage fails.

2 Apply starting air, ignition, fuel, and generator excitation, and shut off starting air and ignition in the proper sequence.

3 Regulate fuel to the power plant during acceleration in accordance with a fuel flow versus speed relationship which will provide the maximum acceleration possible without producing compressor surge.

4 Govern the power-plant speed after rated rpm is reached and apply load.

5 Shut the power plant down and provide an annunciator signal, should malfunction such as excessive gas temperature, overspeed, or low lube-oil pressure occur.

The control system employs a 24-volt d-c circuit energized by a small storage battery. A Woodward multicontact speed switch, mounted outboard from the fuel pump, supplies signals at 4000, 11,000, 20,000, and 22,000 rpm, thus providing the necessary sequencing. The speed-switch signals, together with those from the lube oil underpressure switch and the exhaust-gas overtemperature switch, are fed into the turbine control box where a series of relays and timing motors provide appropriate controls.

The fuel system consists of a micronic filter, gear-type fuel pump, acceleration control valve, governor, and solenoid-operated shutoff valve preceding the fuel nozzle, Figs. 7 and 8.

The fuel-pump delivery is greater than that required for any condition of operation. During acceleration the excess

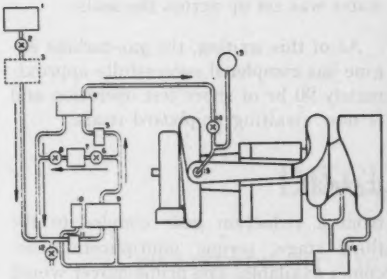


FIG. 7 FUEL-FLOW DIAGRAM
(1, Fuel tank; 2, shutoff valve; 3, boost pump (if used); 4, governor; 5, filter; 6, shutoff valve; 7, acceleration control valve; 8, metering valve; 9, fuel pump; 10, speed sequence switch; 11, filter; 12, drain valve; 13, fuel pressure gage; 14, shutoff valve; 15, fuel inlet; 16, drip tank.)

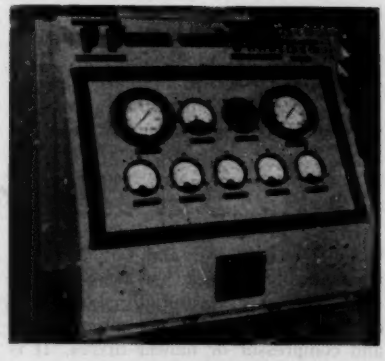


FIG. 8 CONTROL BOX

fuel is by-passed as necessary by the acceleration control valve which senses fuel-nozzle pressure, which in turn is proportional to speed, and thus provides the proper fuel-speed schedule. After rated speed is reached, the acceleration-control valve is removed from the system and a conventional Woodward fly-ball-type isochronous-speed governor by-passes fuel as necessary to maintain speed within the required limits.

SILENCERS

The principal source of objectionable noise in a shaft-output engine of this type lies in the compressor rotor blades and stator vanes. The level of high-frequency sound emanating from the compressor is reduced by a circular muff-type silencer fitted around the compressor inlet. Two concentric rings or splitters, made up of Fiberglas insulation encased in perforated aluminum sheet, are located in the air stream passing through the silencer. The silencer inner walls are insulated similarly. Lower-frequency sound in the turbine exhaust is reduced in intensity by means of a muffler designed along the lines of conventional reciprocating-engine muffler practice. Enlarged flow areas minimize pressure loss with the high volumetric flow passing through the muffler.

TEST PROGRAM

The test operation time accumulated prior to delivery served to point up and permit the correction of operational faults of the type discernible after short-time operation. However, it is recognized that the endurance properties of the equipment are not yet proved, particularly properties affected by factors such as vibration, thermal fatigue, stress rupture, and creep phenomena. The simple and to a large degree conventional mechanical design paid ample dividends in the remarkably trouble-free mechanical operation. Differential thermal expansion arising from the rapid starts and shutdowns resulted in various seal rubs; and minimum clearances had to be established largely on a cut-and-try

basis. Replacement of the single labyrinth oil seals with double pressurized seals was found to be desirable. This eliminated the escape of lubricating oil in the form of oil mist when a pressure difference of more than a few inches of water was set up across the seals.

As of this writing, the gas-turbine engine has completed successfully approximately 90 hr of shore test operation and is now awaiting shipboard trials.

OTHER APPLICATIONS

In its present design form with the turbine, compressor, and output shafts mechanically coupled, the prime mover is suitable for substantially fixed-speed applications such as generator, pump, and compressor or blower drives. It is particularly suitable where use factors are low and space and weight are at a premium. By adding a third turbine stage not mechanically connected to the first two, and taking the power output from a reduction gear coupled to the third stage, torque multiplication becomes available. The prime mover would then be suitable for propulsion of land vehicles or small boats, and other variable-speed applications. Improvement in turbine efficiency, resulting in increased power output and improved economy, also would be associated with the use of three turbine stages.

DISCUSSION

BY LIEUTENANT (JG) MERSON BOOTH, USN,

of the article "The Education of Naval Engineers" by Captain Guy Chadwick, USN, Ret., appearing in the February 1952 issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

THE AUTHOR

A graduate of the Naval Academy in the class of 1947; served as a general line officer in cruisers; assigned to the Armed Forces Special Weapons Project, Sandia Base, Albuquerque and Los Alamos, New Mexico; selected for post-graduate instruction in Naval Construction and Engineering at the Massachusetts Institute of Technology in 1950.

The author has obviously given this matter of professional education considerable thought. I think sooner or later (usually too late) a student engineer will ask himself—what are the objectives of this education he is pursuing. And I think that when he does ask himself this question he most likely answers it with the nebulous thought that he is preparing to be a successful practicing engineer. The definition of what encompasses this aim is difficult for the student engineer to make. I feel that the author's discussion of the ingredients which make up a successful Naval Engineer, encompassing: a high development of skills, a large store of retained facts, the power of reasoning to apply these skills and facts to the nature of a ship, are a well considered delineation of the professional attributes. I have nothing constructive to add to this.

However, in extending the objectives of success in the profession to the formal education curriculum, I feel that the author has made a serious error of omission, in words of the article, if not in thought. And this omission may have resulted in a false sense of adequacy of the curriculum. This error is that the education process, consisting (in the author's division) of skills, facts and reasoning power, is not a transient thing which exists only during the years of formal education, but is a continuing process of development through the life of the individual. Many practicing engineers today are concerned with problems in fields which did not even exist when they went to school. It is safe to say that many of the problems which today's student engineer will encounter in his future career do not even exist today. Thus the curriculum which the

student engineer pursues today must prepare him for problems created by hazily discernible future events.

To develop requisite skills, facts and reasoning power to cope with these future problems is the individual's responsibility after he leaves the educational institution. This self-educating process requires keeping up with technological changes as they occur. And the engineer's future success is determined primarily by how well equipped he is to continue this self-education process, and not primarily by how well he can perform his duties in the profession at its present state of development immediately after graduation. Most engineers will tell you that practically all they know they learned after they got out of school.

I think it is clear then, that the curriculum of formal education be directed to this aim of equipping the individual engineer for self-education through the rest of his life. This means that his curriculum should contain primarily basic science courses (mathematics, physics, chemistry) and applied science courses (mechanics, thermodynamics, fluid mechanics, circuit theory, metallurgy). Applied engineering courses (structures, design of midships' section, hydraulics, etc.) should be given with due regard to their permanency of value to the individual in light of rapid technological advances. Their aim should be to demonstrate to the student how the science he has learned is applied to engineering and to develop in the individual a critical attitude to perceive the problems in applying science to engineering.

I feel that the basic science courses should be of primary importance in the curriculum. The amount of mathematics or physics taken should be in excess of that needed to understand other courses in fluid mechanics or circuit theory. And applied science courses should be taken in excess of the requirement that they have some application to applied engineering courses given.

I make this requirement since the basic and applied sciences are the fundamentals upon which further self-education is possible. To continue self-education in thermodynamics, fluid mechanics, or electronics, to name a few fields, will certainly require more advanced mathematics and physics courses than are now contained in the present Naval Engineering curriculum. Basic science courses are more easily taught and learned in the environment of an educational institution. And a good background knowledge of basic and applied sciences gained in school in excess of immediate requirements, will, after graduation, greatly facilitate and promote initiative in continuing self-education. In short, the courses to be stressed are the ones which will have the most permanent value to the engineer. In these times of great technological advance the order of permanence of value of the various courses is: basic science, applied science, and least, applied engineering.

Perhaps I have been a little too strong in my thoughts on this subject. I have, in effect, recommended that the engineer take substantially the same courses as the scientist. But there is a subtle difference between the engineer and the scientist. The engineer combines both science and the art of applying science to nature. This art is the distinction between the scientist and engineer. This art may be an inborn characteristic of a good engineer, but it certainly should be fostered in engineering school curriculum. And therein should lie the difference between the scientist's and engineer's formal school education. And I don't believe that most of the applied engineering courses, as they are given today foster the development of this art in the individual student engineer. Applied engineering courses are valuable only if they attempt to foster this art

by demonstrating how science which has been learned is applied to engineering and let the student perceive the problems encountered in an applied field which is not the precise field he has encountered in the sciences. Most of the applied engineering courses given today do not accomplish this aim. They are in reality vocational or "Handbook" courses which limit or prohibit individual initiative. They say "This is the way we do it now" instead of "This is why we do it this way; how can we improve it?"

The corrective measures to take are obvious. Revise applied engineering courses to foster this aim. This can best be done by putting the course on a seminar basis with individual or group projects to promote individual realization of problems involved in applied engineering. In addition, practicing engineers can be invited to describe to the students problems which have come up in their fields and how they went about solving them.

Or if there is a feeling (and evidently there is) that this specialized art of engineering practice is better obtained through experience after graduation and since a background of fundamentals is exceedingly difficult to acquire after graduation, I submit that the purely vocational or "Handbook" courses in the curriculum be compressed to a point commensurate to their permanent value to the engineer, and the basic and applied science courses in the curriculum be expanded accordingly.

Evidently there has been a concerted effort made in the present Naval Engineering curriculum to balance the three groups of courses to make a well rounded, complete package of education, while the more important aim of fostering future self-education is ignored. Upon graduation the student should not feel that his education is now complete, well rounded and final, but the educational institution should have indoctrinated into him a feeling of curiosity, initiative, and ability to continue his education process.

BOOK REVIEW

MECHANICS OF VIBRATION

By H. M. Hansen and P. F. Chenea

vii + 417 pp., Illus., 1952, Wiley, New York; Chapman and Hall, London; $8.00

Reviewed by Lieut. Commander S. R. Heller, Jr., USN

This new text fills a void by furnishing teaching material for a first course in Vibration Analysis without recourse to the more scholarly technique of the classic works of Timoshenko, Den Hartog, and others. This lack of scholarly approach is by no means a categoric condemnation of the work; rather, its simple lucid style should be provocative of rather widespread adoption for an undergraduate course.

A single chapter is devoted to a series of definitions and fundamental concepts—an admirable step in preparing new students for a new course. It is, however, unfortunate, in this reviewer's opinion, that the complex variable and vector notation are not also introduced at this point instead of being included in a later chapter.

The first portion is devoted to steady-state vibration of systems of a single degree of freedom—the basis of all analysis. The authors have covered this fundamental theory in clear and meticulous detail. The introduction of relaxation frequency and the energy method and their physical meaning at this early stage furnishes a solid foundation on which the balance of the text is based.

In Part 2 the theory is extended to several degrees of freedom. Initially the discussion centers on the classical analysis including Lagrange's equations. Next, the "mobility" method is introduced and developed by comparing solutions thus obtained with those previously obtained by the conventional classic methods. Once confidence is established in this new tool, the authors extend its use to more complex problems. Finally techniques for solving the general frequency equation are discussed with emphasis on the Holzer and Graeffe methods.

The last portion of the text is included to show the special problems of vibration analysis: distributed physical constants, transients, and non-linear vibrations. The material selected is designed to titillate the serious student with a mathematical urge. It is doubtful if more than token coverage can be obtained in a single semester, but the material presented is undoubtedly helpful for future reference.

In retrospect, it seems advisable to have those portions of the text which can not be covered in a single semester of instruction in small print to guide instructors who will teach from the text, but who do not have the backlog of experience with this material that the authors have. The inclusion of both problems and answers as amplification of the text enhances its value. Although the book is designed, according to the preface, for acceptance by the graduate student, in addition to its normal use by undergraduates, many fine contributions to this branch of mechanics have been neglected with the consequence that there is virtually no bibliography for the interested student to pursue. Many items of critical interest are immersed in the text and consequently will be missed by the inexperienced reader. In spite of these defects the authors have done a commendable job in compressing the theory and techniques of this broad field of mechanics into such a utilitarian text on the level of the undergraduate.

LONGEVITY OF MEMBERSHIP
CORRECTION

Partly due to the fact that the Society's available financial records do not go back prior to 1902, and partly because examination of records of present Honorary Members was overlooked, several errors occurred in the statement in the May, 1952, JOURNAL relative to Longevity of Membership.

It is most particularly regretted that we omitted the name of Professor W. F. Durand, who should have been placed first by many years on the list of living members. Professor Durand was first listed as a member in the JOURNAL for February, 1891, no less than 61 years ago!!! A truly remarkable record. He made many very valuable contributions to our JOURNAL, including the Prize Essay for the year 1898, for which he was awarded a gold medal and Honorary Membership.

In addition to Professor Durand, the following, who have been members for 40 years or more, should have been included:

Robinson, S. M., Honorary Member—43 years.
Johnson, A. W., Honorary Member—42 years.
Robison, S. S., Honorary Member—42 years.
Johnson, H. F., Honorary Member—41 years.

It is with genuine regret that the Society reports that it has learned since the publication of the May, 1952, Journal of the following deaths among its members:

FORTIER, HENRY J., Civil Member
GELHAUS, HENRY F., Naval Member
KLINTMAN, PEHR C., Naval Member
VANCE, LOUIS G., Civil Member

It is with a great deal of pleasure that the Society announces that the following have been received as members since the publication of the May, 1952 issue of the JOURNAL.

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MacDonald, G. Dana, Lieutenant, USNR,
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Pacific, Hiram Roosevelt, Lieutenant, USNR,
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Roach, Richard T., Ensign, USNR,
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2160 W. 106th St., Cleveland, Ohio

TRANSFERRED FROM ASSOCIATE TO CIVIL MEMBERSHIP

Melton, James Steven

RESIGNED

NAVAL

Draim, Frederick W.

ASSOCIATE

Lawson, Comdr. (E) F. W., R.N.

REINSTATED

Cascini, Ernest, A., Naval Member

Goldstein, Marton L., Naval Member

ASSOCIATION NOTES

ORIGINAL ARTICLES

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $40.00 to $150.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

Manuscripts should be mailed addressed to:

Secretary-Treasurer
American Society of Naval Engineers
605 F St., N. W.
Washington 4, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest JOURNAL which has not yet been closed (60 days before publication) and for which insufficient material is already on hand.

Subject Matter and Authors

To assist the editor in programming future issues, the help of all members is solicited. To make it easy, the last page in this issue is readily detachable as you will find. This page has blanks for two entries.

(a) What subject would you, as a member, like to see covered by an article in the JOURNAL.

and

(b) Who do you consider best qualified to prepare such an article authoritatively, and what is his address (if you know).

Any of these forms which are filled in and mailed to the Society will be received with thanks and the editor will follow up to get the suggested articles prepared.

PHOTOGRAPHS

Photographs of current or historical interest to readers of the JOURNAL are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

NOTICE OF DEATHS

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

SOCIETY LAPEL BUTTONS

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold background.

It is available to all members at fifty cents (50c) each.

LIFE MEMBERSHIPS

Any member may now purchase a life membership under the following rules:

(a) Any dues paid for the year in which life membership is purchased will be credited to the cost of the life membership.

(b) Life memberships are non-transferable and terminate with the death of the member.

(c) No refund will be made on account of death or resignation of a life member.

(d) Life membership will vary in cost with the age of the member at his next birthday following application, as follows:

Over 55 years	$125.00
40 to 55 years	175.00
25 to 39 years	225.00
Under 25 years	250.00

(e) All life memberships paid in shall be protected by a transfer on the books of the Society of Series G, U. S. Government 2½ per cent bonds. Thus a life membership will be established. It will be credited with moneys paid in for life memberships and debited with the regular amount of annual dues for each life member each year.

LOCAL CHAPTERS

Local Chapters of the American Society of Naval Engineers may now be authorized by the Council. The following rules for recognition of Local Chapters have been adopted by the Council:

(a) Twenty or more regular members, Naval or Civilian, may apply for a Charter as a Local Chapter.

(b) Each application must be accompanied by a list of members and a copy of local By-laws. The latter must meet the following minimum requirements.

1. All financial dealings of the local Chapter must be independent of the Society and can in no way obligate the Society as a whole. The local chapter shall set up on a non-profit basis. All accounting shall be local.

2. Each active local Chapter shall receive from the Society a grant of $0.50 per year for each naval member, civil member or associate who resides in the local area and who associates himself with the local Chapter. This grant may be used by the local Chapter to defray any necessary expenses, including (a) Letterheads, (b) Notice cards, (c) Postage, (d) Minimum rental allowance, (e) Secretarial service (part time).

3. Any technical papers presented at local chapter meetings, shall be submitted to the Secretary-Treasurer of the Society for consideration for publication in the JOURNAL. Any paper accepted will be paid for at regular rates.

4. Each local chapter shall submit a quarterly report of its activities to the Secretary-Treasurer. These reports must reach Washington, D. C. prior to 15 January, 15 April, 15 July and 15 October. This report shall contain a list of active local members as of the first day of the quarter for the purpose of computing the amount of grant.

5. Associate members of the Society shall be eligible for membership in a Local Chapter, entitled to all the privileges of other members except voting and holding office.

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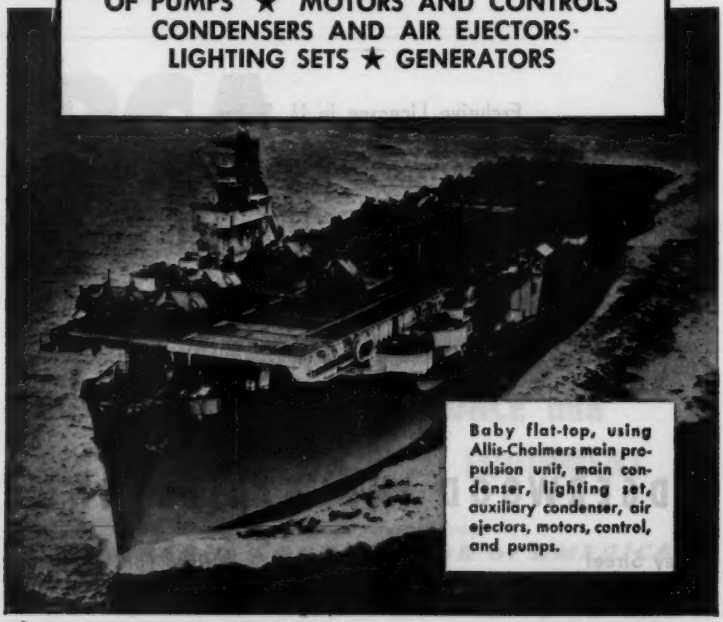

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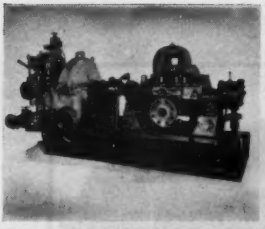

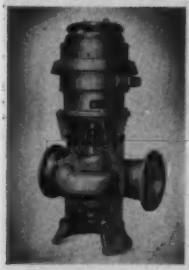
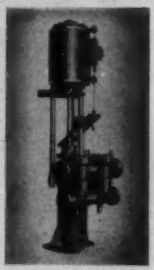
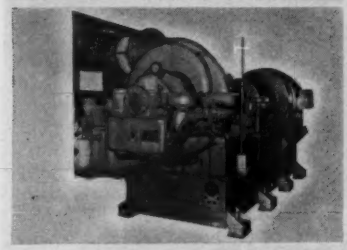
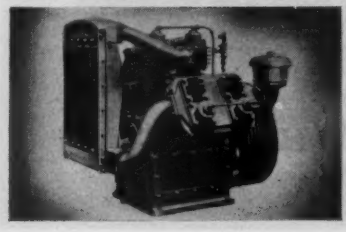

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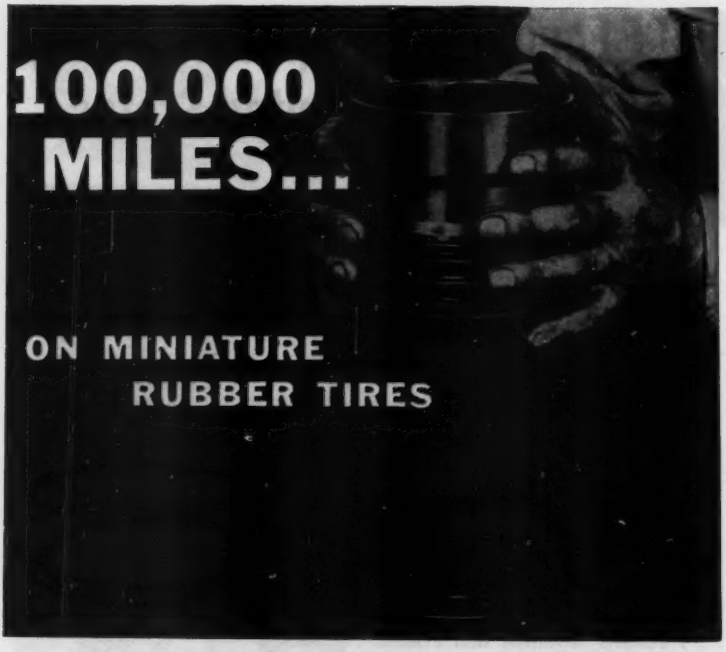

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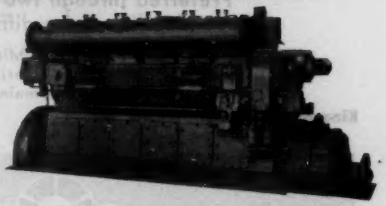

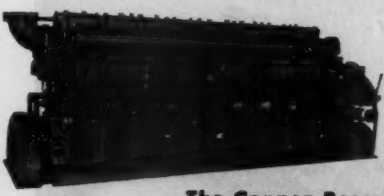

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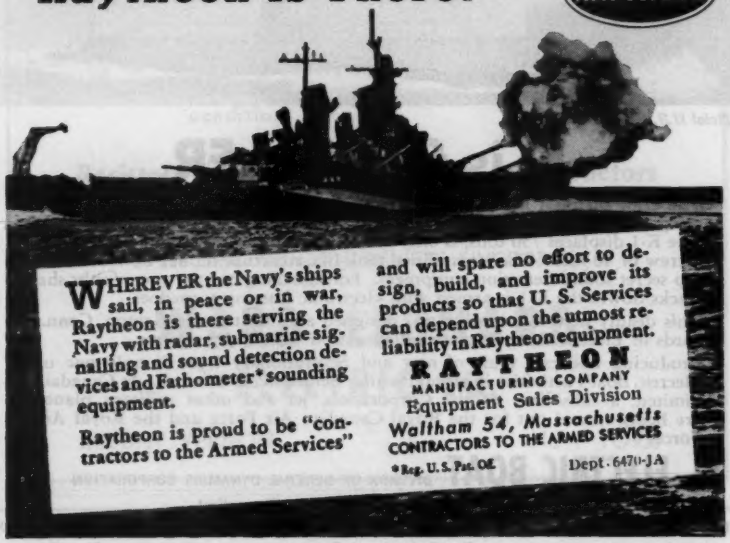

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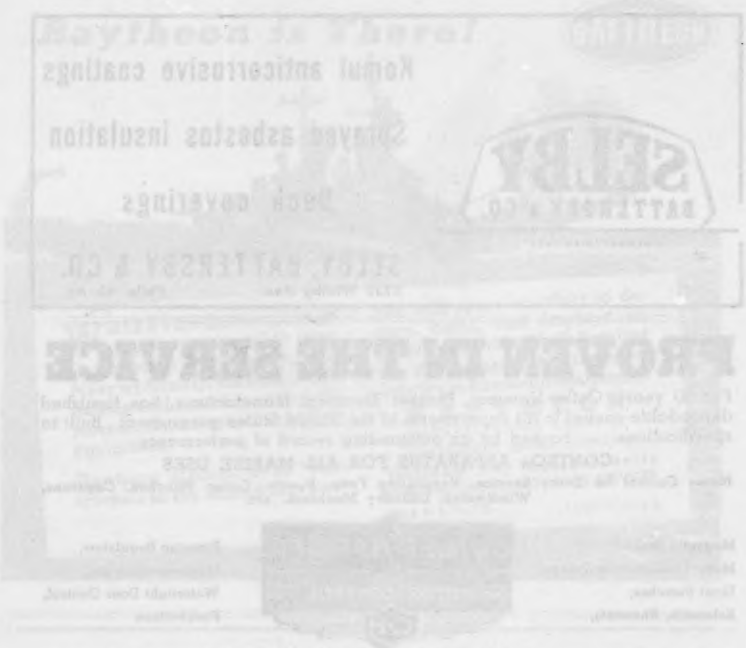

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Persons in civil life who are not eligible for civil membership, but who are especially interested in naval matters or the merchant marine may be eligible as associate members. Commissioned officers of the United States Army and of foreign military and naval services may be eligible as associate members. Persons eligible to associate membership may be admitted upon application and payment of annual dues, provided the application have the recommendation of a member and provided the application shall receive the approval of a majority of the Council, except that in the case of commissioned officers of the United States Army and of foreign naval and military services, the recommendation of a member will not be required.

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Recommended by (two members)_____

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Signature of Applicant

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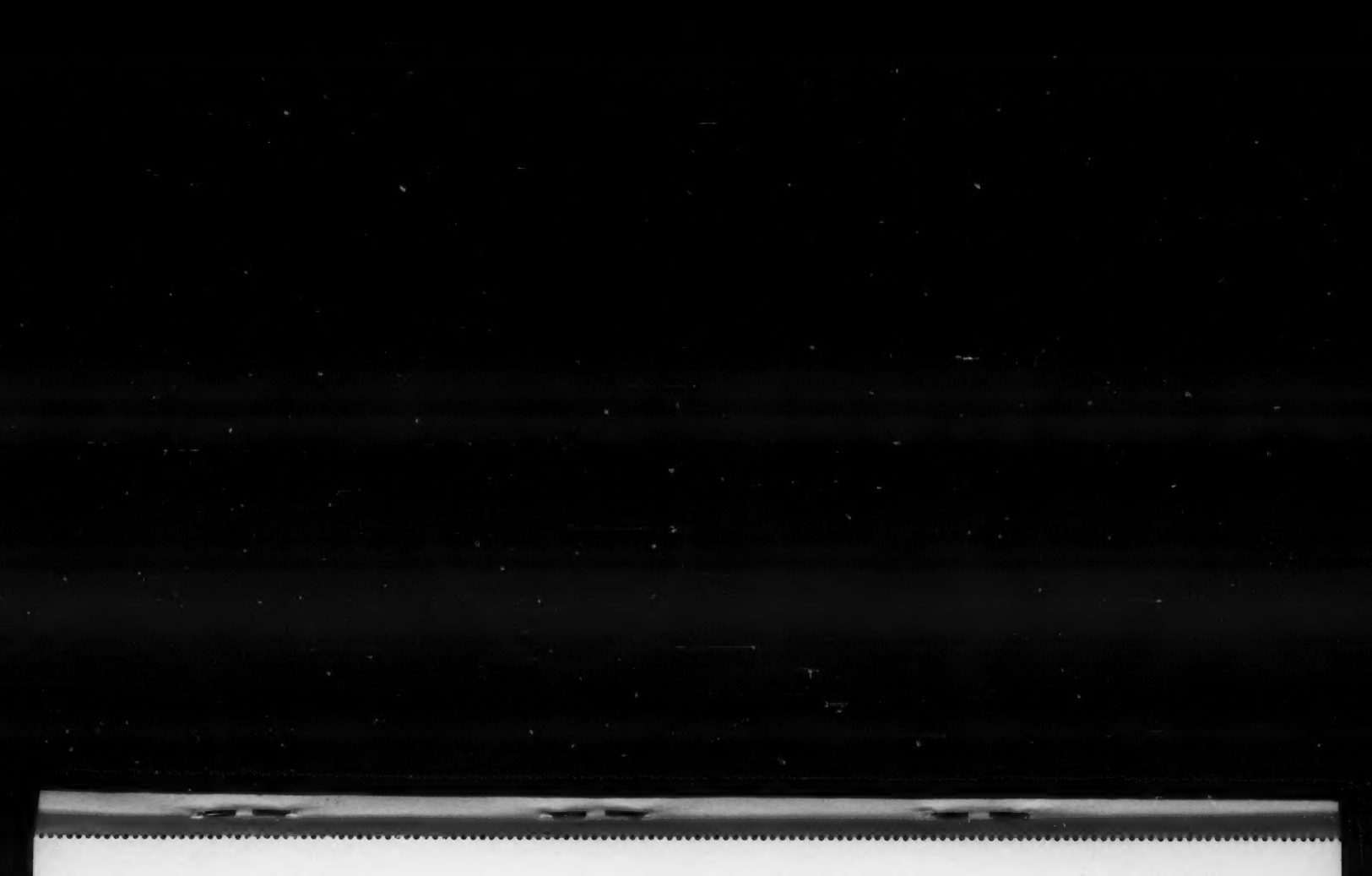
*See reverse side for required qualifications for various classes of membership



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